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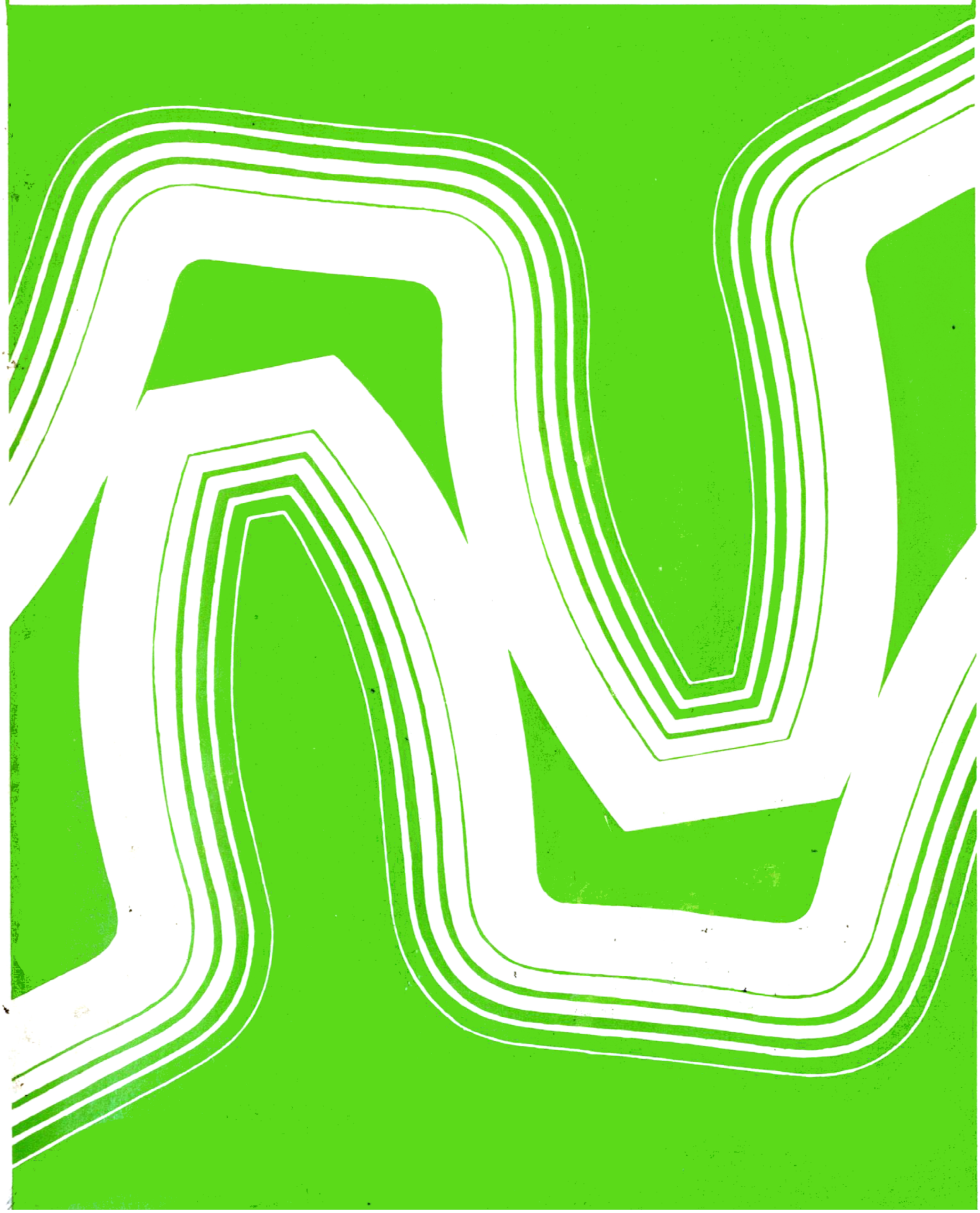
FUNDAMENTALS OF MACHINE DESIGN

MIR PUBLISHERS MOSCOW

P. ORLOV



A technical and methodic handbook for the designers of machines, useful as a textbook for mechanical engineering faculties. Gives a systematic exposition of the rules for rational machine design on the basis of unification and standardization, and for building industrial machines incorporating in their design provisions for further development, lengthening of service life, and improvement of reliability. Surveys ways of reducing weight and increasing strength and rigidity. Considers problems of thermal stress and the effect of thermal deformation on the operation of parts and sub-assemblies. Describes methods of improving efficiency, increasing sturdiness, making assembly easier, and improving operating conditions.

ИЗДАТЕЛЬСТВО МИР

П. И. ОРЛОВ

ОСНОВЫ КОНСТРУИРОВАНИЯ

ИЗДАТЕЛЬСТВО
"МАШИНОСТРОЕНИЕ"
МОСКВА

3 FUNDAMENTALS OF MACHINE DESIGN

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The Greek Alphabet

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ ϰ	Kappa	Σ σ	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	ϒ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ ϑ	Theta	Π π	Pi	Ω ω	Omega

The Russian Alphabet

А а	a	К к	k	Х х	kh
Б б	b	Л л	l	Ц ц	ts
В в	v	М м	m	Ч ч	ch
Г г	g	Н н	n	Ш ш	sh
Д д	d	О о	o	Щ щ	shch
Е е	e	П п	p	Ъ	’’
Ё ё	e	Р р	r	Ы	y
Ж ж	zh	С с	s	Ь	’
З з	z	Т т	t	Э э	e
И и	i	У у	u	Ю ю	yu
Й й	y	Ф ф	f	Я я	ya

На английском языке

Contents

Chapter 1

Assembly

To provide for efficient and high-quality assembling, the design of connections, units and assemblies must satisfy the following conditions:

(1) full interchangeability of parts and units;

(2) elimination of the fitting together of parts during assembly;

(3) easy access for fitter's tools; the possibility for using power tools;

(4) principle of unitized assembly, i.e., connection of parts into primary subunits, subunits into units and units into assemblies, and mounting the assemblies on the machine.

Should these requirements be fulfilled, the manufacturing process can be organized along the lines of parallel and simultaneous performance of operations, a cycle of constantly repeated operations allotted to each workplace and the process of assembly mechanized. In large-lot and mass production progressive assembly can be effected, if these requirements are met.

The interchangeability of parts is ensured by specifying proper tolerances and limiting form deviations (nonparallelism, nonperpendicularity, etc.). The grades of fit are selected depending on the conditions in which connections are to operate. The required grade of accuracy is established by a dimensional analysis which verifies the operating ability of the joint when clearances (interferences) in the joint have extreme values.

Sometimes operating conditions require that clearances (interferences) be maintained within narrower limits than those obtainable when the mating parts are machined even to the first grade of accuracy.

Thus, joints assembled by heavy drive fits to ordinary grades of accuracy will not be strong enough in the event of unfavourable combinations of sizes (holes machined to the maximum plus tolerance and shafts, to the minimum plus tolerance). In the reverse case (holes machined to the nominal size and shafts, to the maximum plus tolerance) excessive stresses arise in the parts being connected.

When a pin is inserted into piston made of an aluminium alloy, the initial (cold) clearance between the pin and the bosses of the piston sharply increases at the working temperatures due to the high linear expansion coefficient of

aluminium alloys, which may cause damage to the joint. This makes it necessary to fit the pins into the holes of the bosses with an initial interference which disappears with the heating of the piston and is replaced by a clearance of the required size. Calculations show that such a narrow field of tolerances must be adopted for the diameters of the pin and holes that can hardly be obtained even with the first grade of accuracy.

In such cases *selective assembly* is often employed. Parts are divided into several groups depending on the amount of deviation from the nominal size of the part. During assembly, it is common practice to connect only the parts of such groups which in combination with each other provide the required amount of clearance (interference). It stands to reason that the principle of interchangeability is violated in this case. The need for breaking in advance the parts into dimensional groups complicates and retards the production process.

For joints of this kind it is expedient to introduce a higher (precision or zero) grade of accuracy. The present-day methods of finish machining (precision grinding of shafts, broaching and honing of holes) allow dimensional accuracies of 0.5-1 μm to be obtained, which is enough for the joints which are now assembled by the selective method. The higher costs of machining would be compensated for by the simpler and cheaper assembly.

Much attention should be given to the elimination of matching and finishing operations during assembly, and the mounting of parts and units in position with individual adjustment of their mutual arrangement. Matching means bench fitting or additional machining, which slows down the assembly, impairs its quality and makes the parts no longer interchangeable. Matching operations are, as a rule, very laborious. They require a preliminary, sometimes repeated, assembly of units, measurements, testing the functioning of each unit, and subsequent disassembly to introduce the necessary corrections. Every disassembly-assembly operation also involves the washing of parts.

Parts in a correctly designed unit should be made to such an accuracy as would provide for the unit assembled from any components supplied from the finished parts store to be capable of operation. The position of parts in a unit, and of units in an assembly or a machine should be determined by locating surfaces and elements machined in advance.

Even today manual operations, such as the lapping of parts in the joints where high tightness is required (fitting of taper valves, plug cocks, flat distributor slide valves, plungers and cylindrical slide valves, etc.) are made use of when assembling some joints. Lapping is also utilized for heavily loaded taper joints to ensure a close contact and prevent the cold hardening and breaking of the seating surfaces. Lapped parts are not interchangeable because they are lapped in pairs.

However, here too, manual operations can be mechanized not only at the preliminary but also at the final stages of machining. For example, the laborious operation of lapping together the flat surfaces of metal-to-metal joints is replaced by a mechanical lapping of each surface to a standard plate to make the mating parts interchangeable.

1.1. Axial and Radial Assembly

The assembly pattern of a unit largely affects its design and operational properties.

For units with longitudinal and transverse axes of symmetry the following two basic assembly patterns may be adopted: *axial assembly* wherein the parts of a unit are joined in the longitudinal (axial) direction, and *radial assembly* wherein the parts are connected in the transverse (radial) direction. With the axial assembly the jointing planes are perpendicular to the longitudinal axis, and in the case of the radial assembly they pass through this axis.

Figure 1 shows by way of example the assembly of a gear shaft in a housing. The axial assembly is presented in Fig. 1*a*. The housing and its cover, as well as the bearing bushings accommodated therein, are solid. The shaft is inserted into the housing axially and locked by the cover which is centred with respect to the housing by means of a cylindrical shoulder.

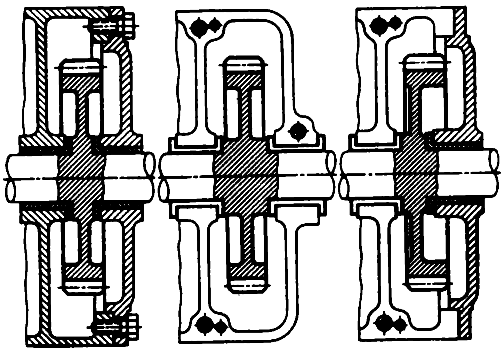


Fig. 1. Assembly of gear shaft into housing

In the case of the radial assembly (Fig. 1*b*) the housing and the bushings are parted along the longitudinal axis. The shaft is fitted into one half of the housing and covered by the other half. Both halves of the housing are located with respect to each other by adjusting pins and clamped by transverse bolts.

Figure 1*c* illustrates a combined radial-axial assembly. In this case the housing is split and the cover, solid.

A multi-step centrifugal pump (Fig. 2) may be taken as an example to show the advantages and shortcomings of the axial and radial assembly.

In the design consistently following the principle of axial assembly (Fig. 2*a*) the housing of the pump is made up of a number of sections carrying diffusors *1* and diaphragms *2* with guide vanes *3*. The unit is assembled by stacking impellers on the shaft (preliminarily

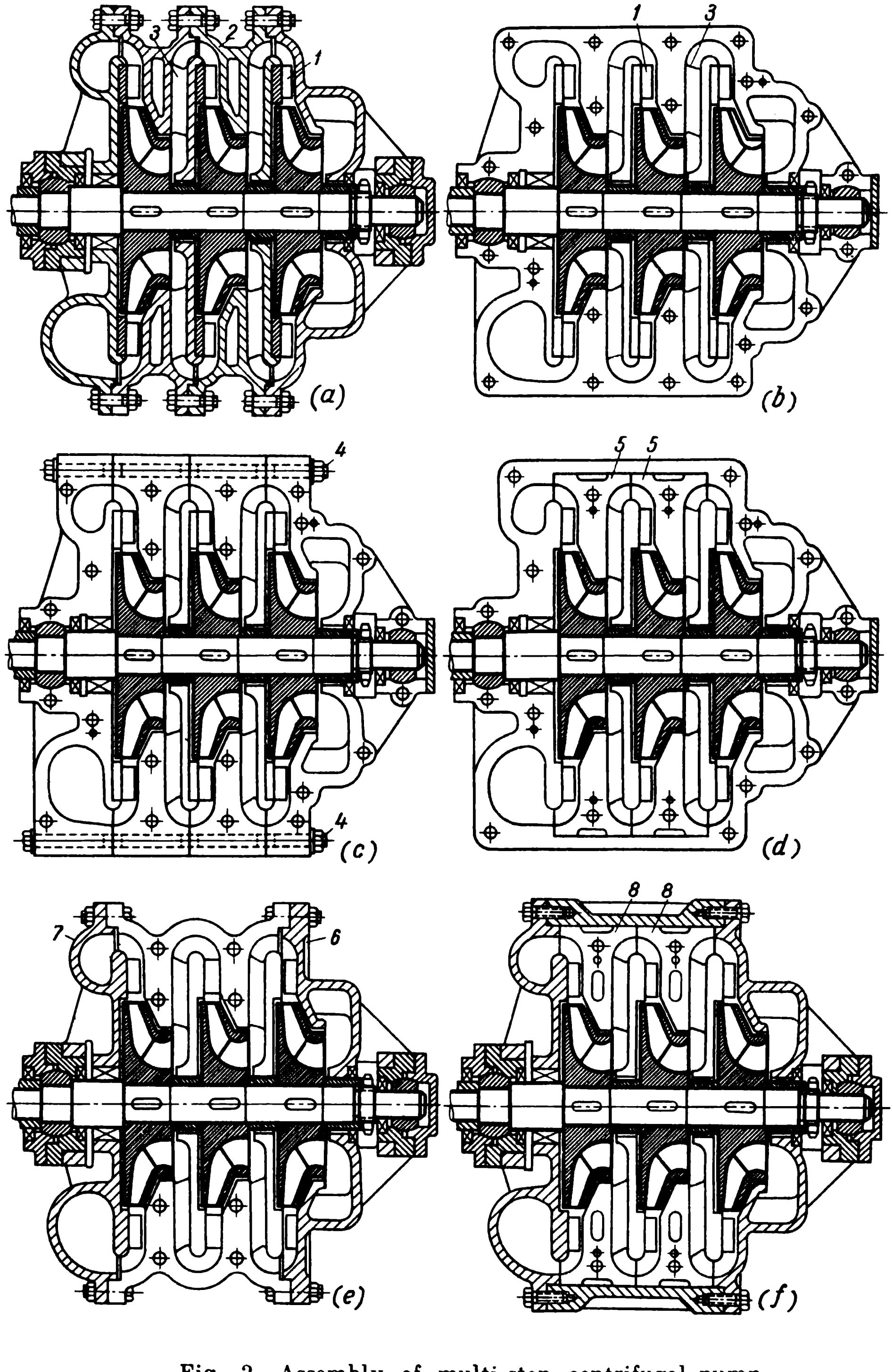


Fig. 2. Assembly of multi-step centrifugal pump

placed into the bearing of the rear cover) consecutively in all the sections, the sections being individually bolted together. The assembly ends with the tightening of the impellers by a nut on the free end of the shaft in the front-cover bearing.

With a purely radial assembly (Fig. 2*b*) the housing consists of two halves parted in the plane of the shaft. The casings of the bearings and guide vanes *3* are cast integral with the pump housing. Diffusors *1* are also parted. The diffusor and guide vanes are butted together in the parting plane of the pump housing. The pump is assembled in the following order. The impellers are stacked and clamped on the shaft, the assembled shaft is installed in the bearings in the lower half of the housing and covered by the other half after which the housing halves are tightened by internal and side bolts.

A comparison of the axial and radial assembly patterns leads us to the following conclusions, common to multi-step units.

In the case of the axial assembly it is easy to cast a sectional housing and its machining is convenient. The surfaces being machined are open to view, accessible for cutting tools, and can easily be measured. Since the machining is performed on continuous cylindrical surfaces, high-speed methods can be employed to make individual compartments.

The design as a whole is highly rigid, and its internal cavities are sealed off well.

The shortcomings of the axial assembly are as follows.

1. Complicated assembly of the unit. It is difficult to check and adjust axial clearances, particularly the face clearances between the impellers and the back surfaces of the diaphragms primarily because the shaft is secured only in one bearing at all assembly stages, including the final stage. Correct clearances can be maintained either by means of special fixtures or by increasing the accuracy of the axial dimensions of the structural elements.

2. Complicated inspection of the internal members, because all the preceding stages have to be dismantled to open any one stage.

The radial assembly is opposite in its advantages and shortcomings to the axial assembly. It is difficult to make the housing comprising two massive castings, and its machining is intricate. The internal cavities are machined either by an open method, i.e., separately in each half of the housing with their subsequent matching, or by a closed method when the halves of the housing are assembled by means of set pins, with the mating surfaces being finish machined earlier. Either method requires special tools, measuring instruments and highly skilled personnel.

Since the housing sections are not symmetric, the housing has unequal rigidity. The rigidity is less in the jointing plane and larger in the direction perpendicular to it. As the structure is weakened by the longitudinal parting, the sections of the housing walls

have to be increased, which makes the unit heavy. The housing cavities are in need of careful sealing along the shaped jointing plane without disturbing the cylindricity of the internal machined surfaces. This is usually attained by lapping together the mating surfaces and using jointing compounds. The diffusor and guide vanes have to be matched in the jointing plane, or one has to use sets of stacked vanes which are individually installed into the annular recesses of the housing.

On the other hand, assembly and disassembly are extremely convenient. During assembly the shaft with the impellers fitted thereon previously is placed into the bearings of the lower half of the housing. The axial clearances can thus be easily measured and properly adjusted. The internal cavities of the unit can likewise be easily inspected. The removal of the upper half of the housing reveals the internal spaces of the unit and provides free access to all the parts installed in the housing.

It follows therefore that the axial assembly is more suitable when a strong and light design is required (transporting machines) and a few operational inconveniences may be allowed. If the mass of the construction is not important and higher manufacturing costs may be allowed to make assembly and operation more convenient, the radial assembly is used.

Various combinations of the elements of the axial and radial assembly patterns are in common use.

In the radial assembly (Fig. 2*c*) to make the casting process easier the housing halves are assembled of separate half-rings clamped by fitted longitudinal bolts *4*. The housing halves thus assembled are machined together on the parting surfaces and further the clamping bolts are not removed. The shortcomings of the design are the increased volume of machining operations and a larger number of butts perpendicular to each other.

In the design shown in Fig. 2*d*, diaphragms *5* are made separately, each of the two halves being bolted together with the use of set pins and fitted into the split housing.

In the combined radial-axial assembly (Fig. 2*e*) the middle portion of the housing consists of two halves that can be detached along the axis of the shaft. Front (*6*) and rear (*7*) covers carrying the bearings are attached to the end faces of the housing. During assembly the shaft with the impellers is placed into the lower housing to which the covers are afterwards attached, and the shaft is centred in the bearings. Then, the upper half of the housing is mounted and the upper bolts of the covers are tightened. During disassembly for inspection the covers remain screwed to the lower half of the housing.

With such a design the manufacture of split housings is simpler and assembly and disassembly are as convenient as before.

In the combined assembly (Fig. 2*f*) each diaphragm *8* is made up of two halves and inserted into the solid housing together with the shaft and the impellers according to the axial assembly principle.

The assembly patterns of a single-step reduction gear in which the axes of the gears are arranged in a horizontal plane are illustrated in Fig. 3.

In the axial assembly (Fig. 3*a*) the presence of the base does not allow the housing to be split along the axis of symmetry. The gears

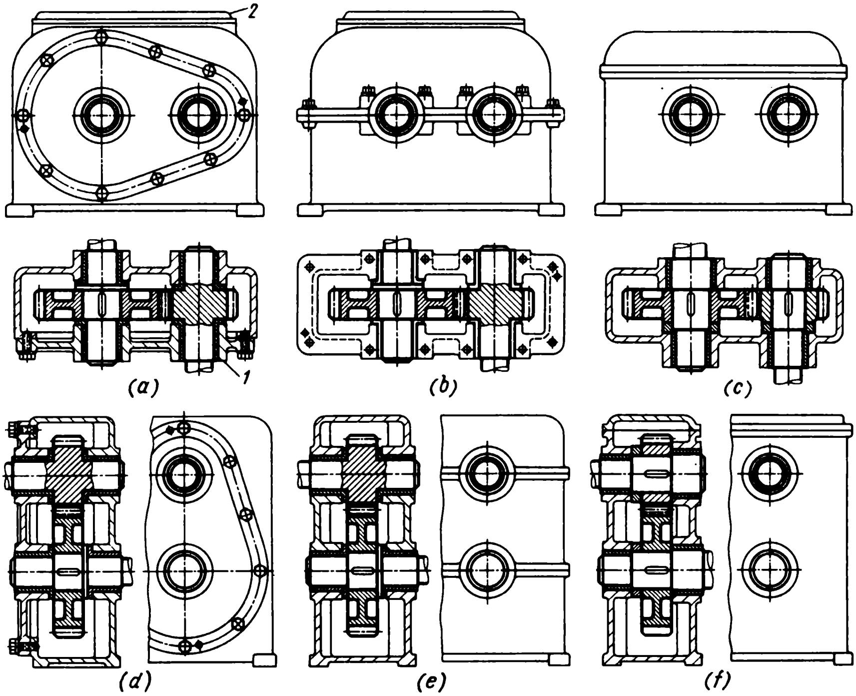


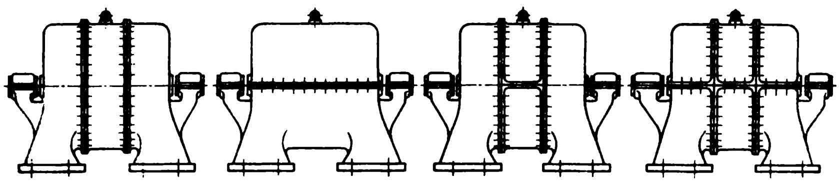
Fig. 3. Assembly of single-step reduction gears

are mounted in the wall of the housing on one side, and on the other in its detachable cover *1* located on the housing by set pins. The design provides for the convenient machining of the housing. As distinct from multi-step units, installation is also convenient. Inspection hole *2* is used to check the meshing of the gears and inspect the interiors of the reduction gear.

n the radial assembly (Fig. 3*b*) the housing consists of two parts oined in the plane of the gear axes, the parts of the housing being ixed with respect to each other by set pins. Like other radial assemblies this design is difficult to machine. The seating holes to receive the shaft bearings are machined in the assembled housing, the

mating surfaces of the housing halves being machined previously, or individually in both halves with the subsequent finish machining of the jointing surfaces. The latter method is more complicated than the former one.

The sealing of the joint involves some difficulties. Elastic gaskets must not be used lest the cylindricity of the bearing seats should be spoiled. The mating surfaces should be lapped together and sealed with jointing compounds. It is especially difficult to seal off simultaneously the flat joint and the external cylindrical surfaces



(a) (b) (c) (d)

Fig. 4. Detaching the housing of rotory machine

of the bearings (if the bearing bushings are solid). An inspection hole should be provided in the housing lest the joint should be disturbed during operation.

In this case the axial assembly is preferable. It allows easy machining and good installation.

In the combined radial-axial assembly (Fig. 3*c*) the shafts of the gears are supported in the walls of the housing provided with a cover having its parting plane arranged above the bearing seats.

The assembly takes the following course: the gears are introduced into the housing, the shafts are passed through one of the bearings and through the gear hubs (the shafts should be stepped) and the gears are fastened to the shafts. This design is much better than the previous ones because of simpler machining and more stable position of the shafts in the housing though the installation is more difficult.

Figure 3 *d-f* shows a reduction gear with gears arranged in a vertical plane. The axial (Fig. 3*d*), radial (Fig. 3*e*) and radial-axial (Fig. 3*f*) assemblies have respectively the same advantages and shortcomings as the designs shown in Fig. 3*a*, *b* and *c*, the only difference being that the shortcomings of the radial assembly are here more evident due to the presence of two butts.

Sometimes the pattern of assembly is unambiguously defined by the design of the unit. Thus, the axial assembly (Fig. 4*a*) is out of the question in the case of a stationary rotary machine mounted on a foundation, because it would be necessary to remove the machine from its foundation to inspect its internal mechanisms. Only the

radial assembly (Fig. *4b*) or the limitedly combined assembly (Fig. *4c* and *d*) is possible in this case.

It is practically impossible to use the axial assembly for crankshafts of multi-cylinder piston engines because of the shape of the shafts and the installation conditions of the split ends of the connecting rods.

The radial assembly is not always possible for cup-type parts such as impellers (Fig. 5). The design illustrated in Fig. *5a* can be assembled only by the axial method because the radial assembly of the housing is impeded by the projection (by amount *m*) of the impeller disk with respect to the housing hubs.

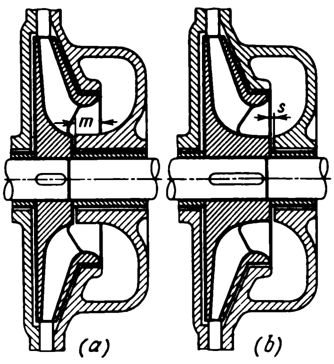


Fig. 5. Assembly of enclosed impeller

For the radial assembly the hub should be shortened (Fig. *5b*) and an axial clearance *s* left between the impeller and the hub.

In most cases several assembly versions may be utilized. The task of the designer is to select the one most suitable for the given conditions of operation.

Let us discuss the methods of the radial and axial assembly of a standard gearbox (Table 1).

All the radial assembly versions (drawings *1-4*) fully ensure unitized assembly, allow convenient gear engagement checking and adjustment of the gear positions with respect to the adjacent parts.

However, manufacture is more complicated. The joint between the housing halves must be thoroughly machined and the seating surfaces and their end-faces machined conjointly in the assembled housing halves. Soft sealing gaskets in the joint must never be used lest the fit of the bearings in their seats should be spoiled. The parting weakens the housing, and its rigidity has to be increased by making the walls thicker, employing ribs, etc. The pattern can only be applied if the axes of the other gears of the drive are also arranged in the parting plane.

The axial assemblies (drawings *5-19*) are more simple to manufacture. The strength and rigidity of housings are as a rule higher. In mechanisms with multiple gears the gear axes may be located in different planes. The centre distance between the adjacent gears is restricted in some designs (drawings *8-11*).

Mounting is more complicated in the systems of axial assembly.

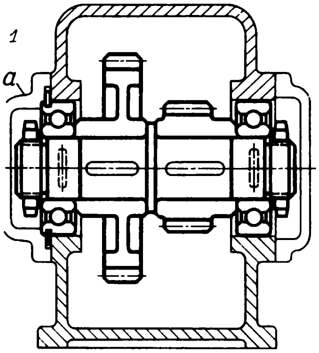
In both systems inspection holes ensure convenient servicing during operation (drawings *2-4*, *8-19*).

Table 1

Cluster Gear Assembly Patterns

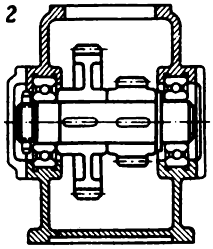
Radial assembly

1



The parting plane of the housing passes through the axis of the cluster. The bearings of the shaft with assembled gears are placed on the seating surfaces of the lower half of the housing and covered by the upper one which is located with respect to the lower half by set pins. The left-hand bearing is fixed by cover *a*, the right-hand bearing is floating.

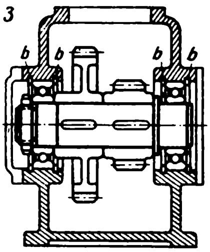
2



The upper half of the housing is located with respect to the lower one by the outer bearing races. The right-hand bearing floats on the shaft.

The shortcoming of this design is that it is impossible to through-pass machine bearing seating surfaces.

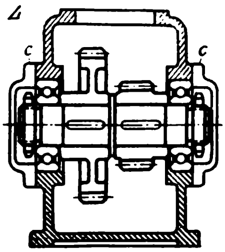
3



The halves of the housing are located with respect to each other by the outer bearing races and rings *b*. The right-hand bearing floats on the shaft.

The bearing seating surfaces can be through-pass machined.

4

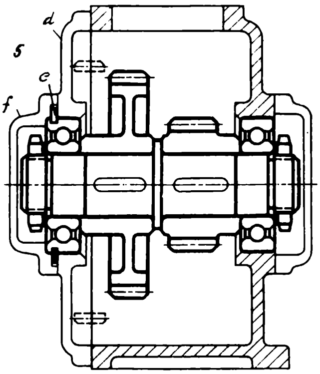


The halves of the housing are located with respect to each other by the bearing races and covers *c*. The design may be applied when the distance between the bearings is not too large.

Table 1 (continued)

Axial assembly

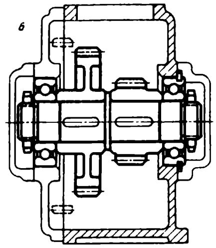
5



The detachable wall *d* is located with respect to the housing by set pins. During assembly the cluster is installed with its right-hand bearing into the housing and covered by the detachable wall (lock ring *e* of the bearing should first be removed) after which the cluster is secured by cover *f*.

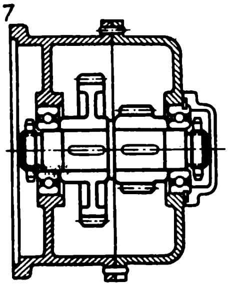
The shortcomings of the design are the reduced rigidity of the housing and the position of the sealing gasket below the oil level.

6



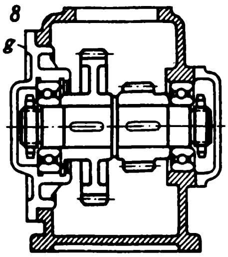
The cluster is secured in the axial direction by the bearing in the housing.

7



Another design version (suspended housing).

8



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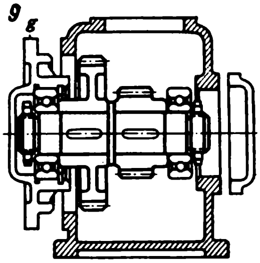

The housing (drawing *8*) has a hole with a diameter exceeding that of the larger gear. The cluster is installed in cover *g* and inserted into the housing (drawing *9*).

The centring surfaces in the housing are machined in one operation.

The diameter of the cover restricts the arrangement of adjacent gears in the gearbox.

Table 1 (continued)

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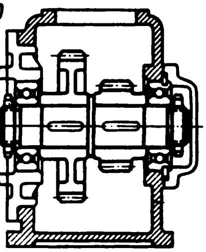

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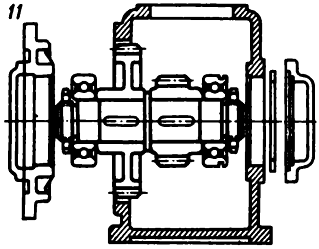
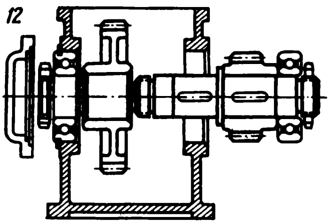


The cluster is fixed by the bearing arranged in the housing. The hole in the cover is intended for the through-pass machining of the seating surfaces.

12



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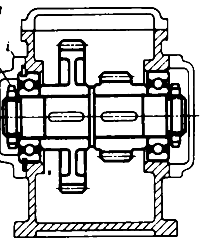
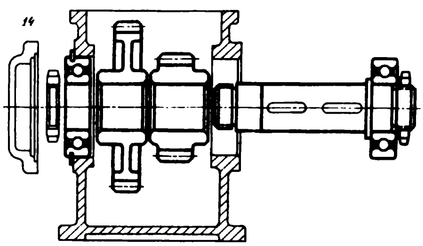


The larger gear is inserted through the upper hole in the housing (drawing *12*) and the shaft carrying the smaller gear is passed through it after which nuts *h* are tightened and the cluster is secured with cover *i* (drawing *13*).

14



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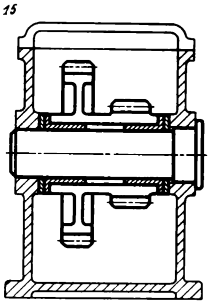
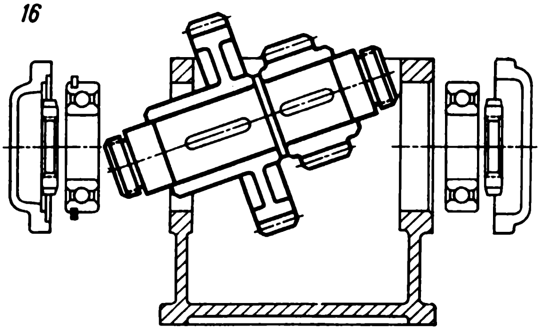


If the diameter of the smaller gear exceeds that of the bearing seat, both gears are inserted into the housing from above (drawing *14*).

When gears are mounted on the shaft on sliding-contact bearings, the assembly is commonly done by passing the shaft through the gears (drawing *15*).

4 *Table 1 (continued)*

16



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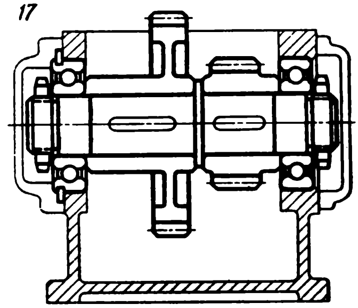
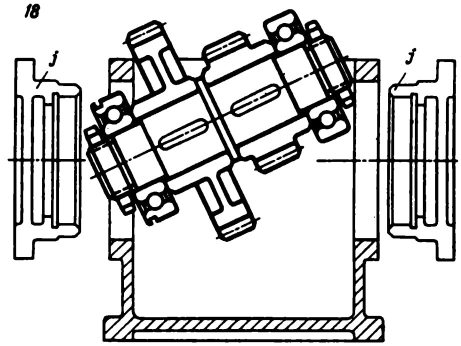


The shaft assembled with the gears is inserted in an inclined position through the upper hole in the housing (drawing *16*) and turned, after which the bearings are mounted and the cluster is secured with the cover (drawing *17*).

18



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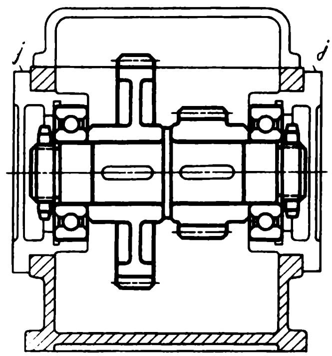


The cluster complete with the bearings can also be assembled by the same method (drawing *18*), if the bearings are mounted in intermediate bushings *j* (drawing *19*) and the upper hole is somewhat enlarged.

1.2. Independent Disassembly

The assembly pattern should be selected so as to ensure a convenient inspection, checking and adjustment of the units. The removal of a part or unit should not disturb the integrity of the other units to be checked.

The gear shown in Fig. 6*a* is obviously mounted unhappily. The gear is locked by nut *1* also serving to fasten the stud shaft in the housing. The entire unit has to be disassembled to remove the gear.

In the improved design (Fig. 6*b*) the shaft and the gear are secured separately, and the gear can be taken off without removing the shaft.

In the fastening unit of a bearing (Fig. 6*c*) the cap and the bottom member are clamped by through bolts. The bearing falls apart as soon as the cap is removed. In the design shown in Fig. 6*d* the cap and the bottom member are disassembled separately.

Figure 6*e* shows a bevel transmission to a camshaft. The bottom members of the bearings are made integral with the frame and the

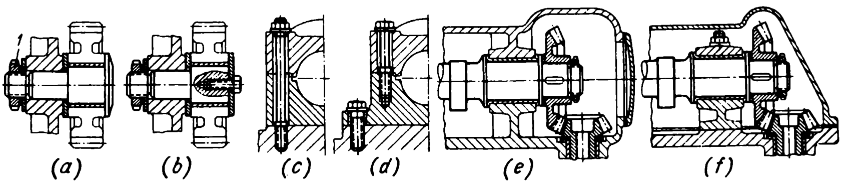


Fig. 6. Assembly patterns

caps form a single whole with the housing of the frame. When the housing is removed, the shaft remains in the lower half-liners, and it is impossible to check the operation of the unit.

It is better to make the housing of the frame independent and fasten each cap to the bearings separately (Fig. 6*f*). After the housing is removed, the entire mechanism is open to inspection. Apart from convenient disassembly, this design makes it easier to accurately machine the bearing holes.

1.3. Successive Assembly

When several parts are successively mounted on a single shaft by an interference fit, one-diameter fits should be avoided (Fig. 7*a*, *c* and *e*). The mounting and dismantling grow in complexity because the parts have to be moved over the seating surface, and there is a hazard of damaging it. In such cases it is more expedient to employ stepped shafts with the diameter of the steps increasing successively in the direction of assembly (Fig. 7*b*, *d* and *f*).

It is especially difficult to assemble a large number of parts on long shafts with a heavy drive fit (Fig. 8*a*). The assembly can be facilitated by heating the parts to be fitted on to a temperature that allows them to be freely mounted on the shaft (although this operation complicates the assembly). This cannot be done during disassembly.

A correct design with a stepped shaft is shown in Fig. 8*b*.

If there are many steps, the standard shaft diameters have to be relinquished and individual dimensions introduced to prevent excessive increases in the diameter of the last steps of the shaft. The

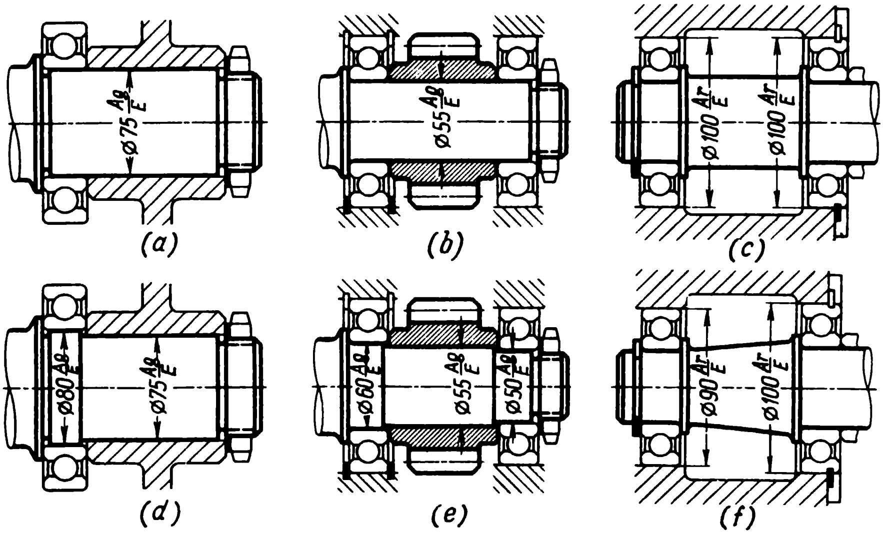


Fig. 7. Assembly with several seating collars
a, c, e—wrong; *b, d, f*—correct

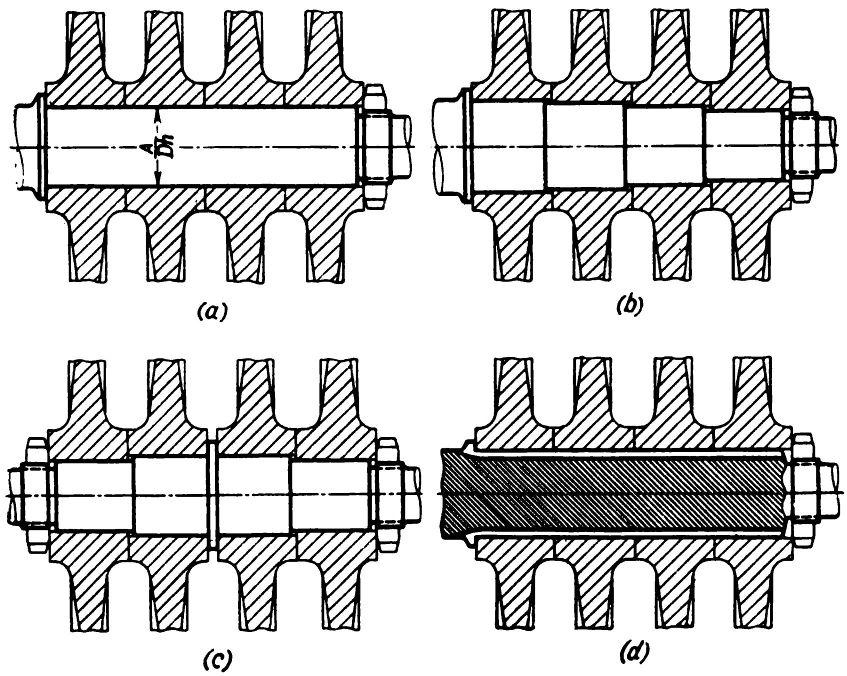


Fig. 8. Fitting of axial compressor disks

difference between the diameters of the steps is reduced in this case to the minimum (about several tenths of a millimetre) enough to fit the parts on easily.

It is better if the assembly is effected from both ends of the shaft (Fig. 8*c*). In this case the shaft and the hubs can be machined much

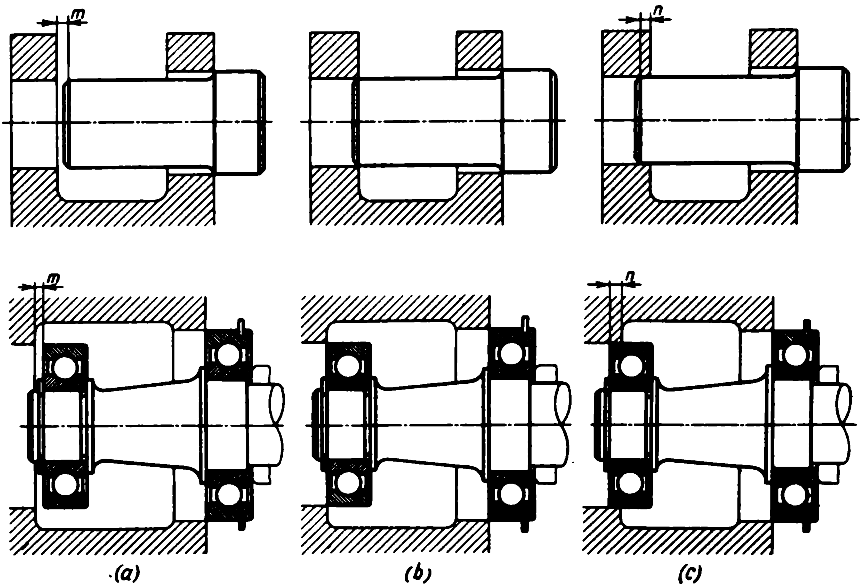


Fig. 9. Installation with two seating surfaces

easier. The number of nominal diameters and the range of special cutting tools (reamers, broaches) and measuring tools (snap and plug limit gauges) are halved.

If parts are mounted on a shaft by a slide or easy slide fit, it is good practice to use a smooth shaft. This also refers to spline-fitted connections (Fig. 8*d*): stepped diameters make the manufacture of the unit much more difficult since each hub requires special broaches, and special hob cutters are needed for each step of the shaft when centring is done from the internal diameter of the splines.

When assembling parts having two seating surfaces, the parts should be fitted into their seats locating in a proper sequence. If the part first fits into the first seat (in the direction of motion) and a clearance *m* (Fig. 9*a*) remains between the end face of the part and the second seat, the inevitable skewing of the part hampers its proper installation, and even makes it altogether impossible when heavy drive fits are employed. All the seating surfaces of a part (Fig. 9*b*) should never come into contact with their mating surfaces simul-

taneously. Correct designs are illustrated in Fig. 9*c*. The part should first fit into the second seat to a distance *n* (2-3 mm) enough to guide it properly, and then into the first seat.

1.4. Withdrawal Facilities

Such facilities must be provided without fail in interference-fitted connections, in connections using sealing compounds or having

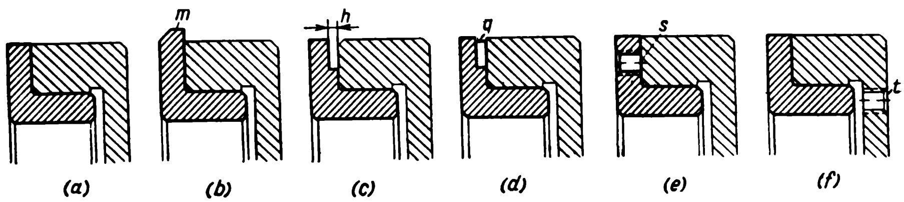


Fig. 10. Withdrawal facilities

parts difficult of access, and also in connections operating under cyclic loads when cold hardening and frictional corrosion may occur.

Disassembly is made much easier if parts are designed with beads, flanges, threaded surfaces and holes, etc.

Figure 10 shows a bushing interference-fitted into a frame. The design shown in Fig. 10*a* is difficult to disassemble. The dismant-

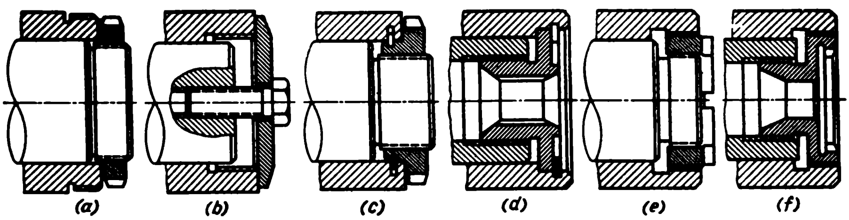


Fig. 11. Withdrawal facilities for tightly fitted hubs

ling process can be facilitated by increasing the height of flange *m* (Fig. 10*b*), by introducing annular clearance *h* (Fig. 10*c*) or recess *q* for a withdrawal tool (Fig. 10*d*) between the flange and the housing, or by providing threaded holes for puller screws either in the bushing (hole *s* in Fig. 10*e*) or in the housing (hole *t* in Fig. 10*f*). At least three threaded holes spaced (at 120°) are necessary to remove the part without skewing.

Figure 11 shows withdrawal facilities employed to pull tightly fitted hubs off cylindrical surfaces.

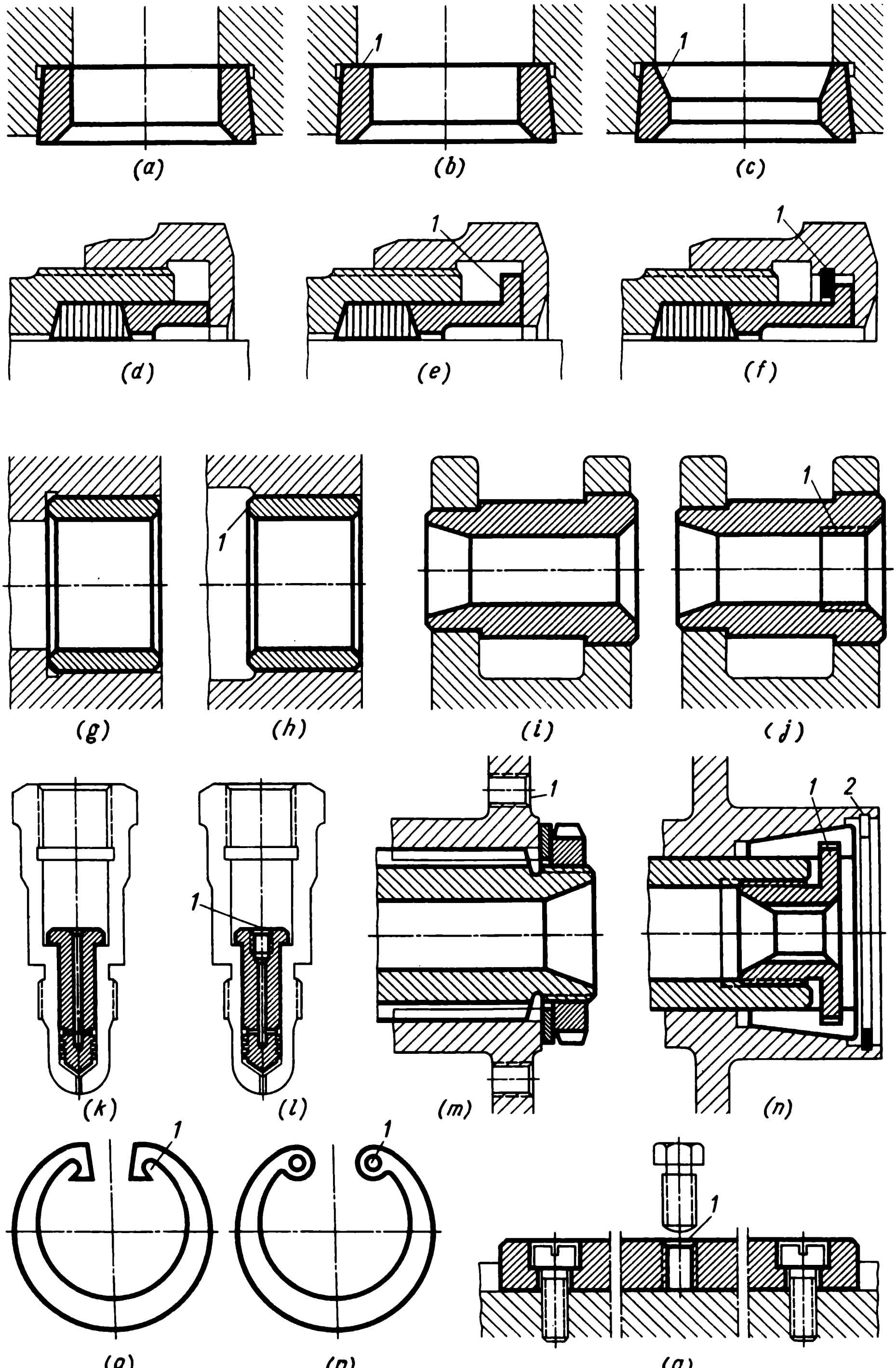


Fig. 12. Withdrawal facilities in standard machine elements

The hub in the designs in Fig. 11 *a* and *b* is provided with a thread for a puller. In Fig. 11*c* and *d* the circlips introduced into the hub serve as pullers.

A system of differential threads is shown in Fig. 11*e* and *f*. The clamping nut has two threaded surfaces each with a different pitch. As the nut is unscrewed, the hub is removed from the shaft.

Figure 12 illustrates some examples of withdrawal facilities (designated by figure *1*) incorporated into the design.

It is practically impossible to replace the press taper-fitted valve seat in the design shown in Fig. 12*a*. The joint can be made detachable if the hole in the body is enlarged with respect to the seat edges (Fig. 12*b*) or the seat is provided with an internal taper (Fig. 12*c*). Then, it becomes possible to press out the seat by applying a force to the top of the seat.

Stuffing-box glands (Fig. 12*d*) frequently jam because the packing is forced into the clearance between the gland and the shaft. A stuck gland can be removed from the box only if a withdrawal means, in the form of a flange (Fig. 12*e*) for example, is provided on the gland.

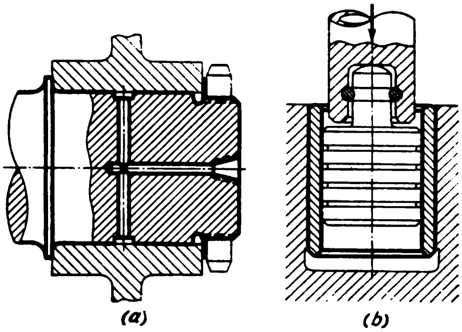


Fig. 13. Hydraulic withdrawal

The best method is to install a lock ring in the gland nut (Fig. 12*f*). In such a design the gland leaves the box as the nut is unscrewed.

A bushing press-fitted into a hollow shaft is shown in Fig. 12*g* and *h*. In Fig. 12*g* the bushing can be pressed out only by damaging it, for example, if a threaded taper rod is screwed into it. In Fig. 12*h* the bushing can be forced out by pressing against its end face.

Other examples of wrong and correct designs are illustrated in Fig. 12*i* and *j* (press-fitting of a pin) and *k* and *l* (installation of a swirler in an injector).

Some methods to ease the dismantling of hubs are presented in Fig. 12*m* and *n*. The hub in Fig. 12*m* is provided with holes for a puller. In Fig. 12*n* (a hub mounted on centring cones) the flange of the clamping nut is inserted into an annular groove in the split cone. When the nut is unscrewed, it first draws out the cone which then thrusts against lock ring *2* and takes off the hub.

Lips (Fig. 12*o*) or holes (Fig. 12*p*) for tongs are used to facilitate the removal of circlips fitted into holes.

Figure 12*q* shows a feather key provided with a tapped hole for a puller screw.

Of late, parts assembled by heavy drive and wringing fits are taken apart by a hydraulic method (Fig. 13*a*) whereby oil at a pressure of 1,500-2,000 kgf/cm^2 is supplied to the mating surfaces.

The hydraulic method of forcing a bushing out of a blind hole is illustrated in Fig. 13*b*. A plunger is introduced into the bushing bore previously filled with oil. When a force produced by a press is applied to the plunger, the pressure developed in the oil layer forces the bushing out of its seat.

1.5. Dismantling of Flanges

Considerable difficulties are frequently met with when disassembling large-diameter flanged joints sealed off by means of gaskets or jointing compounds, or operating at increased temperatures, because the jointing surfaces stick together. The simplest withdrawal means for such flanges are illustrated in Fig. 14. One of the flanges (Fig. 14*a-c*) is provided with projections or recesses (usually three, spaced at 120°) which allow axial forces to be applied to detach the flanges. Figure 14*d-f* shows designs with projections or recesses on both flanges which can be taken apart with a screw driver inserted between the flanges.

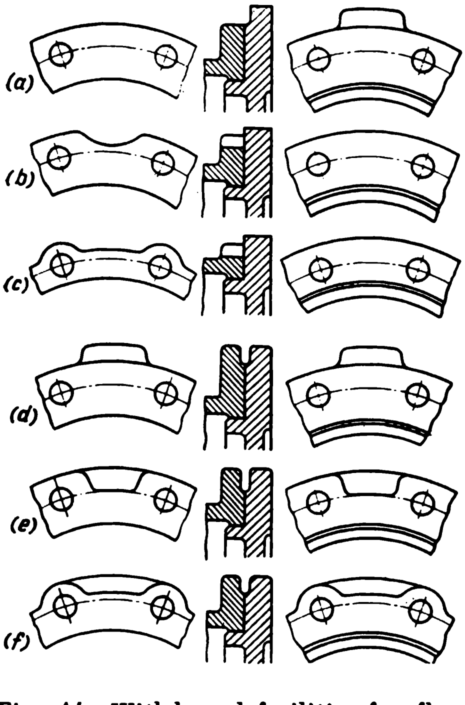


Fig. 14. Withdrawal facilities for flanges

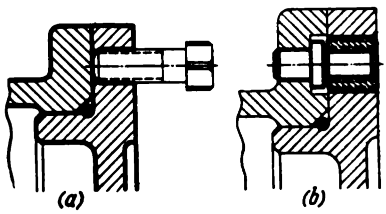
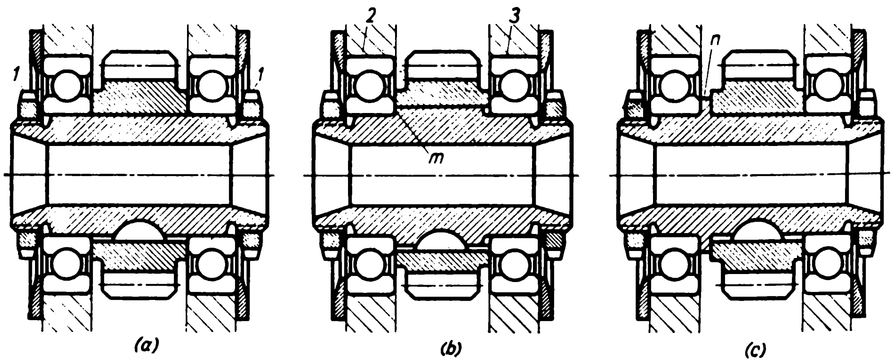


Fig. 15. Separation of flanges by means of pressure bolts

Better withdrawal facilities are presented in Fig. 15. Three threaded holes spaced at 120° are made in one of the flanges. The flanges are separated by means of pressure bolts (Fig. 15*a*) screwed into the holes. The crushing of the jointing surface (especially in parts made of light alloys) is prevented by hardened inserts installed under the pressure bolts (Fig. 15*b*). The holes for the bolts are reinforced with threaded bushings.

1.6. Assembly Locations

The position of parts during assembly should unambiguously be determined by assembly locations. Any uncertainties in design when



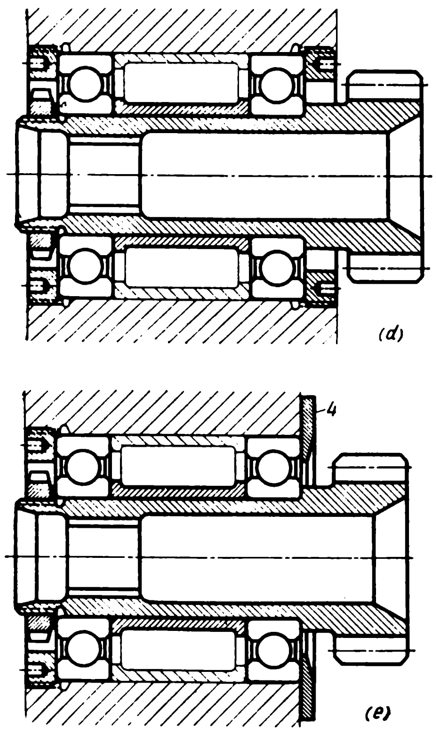


Fig. 16. Locking of parts during assembly

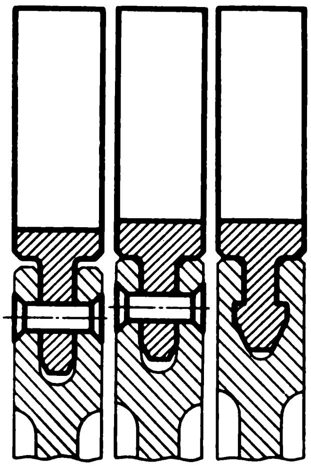
the fitter has to carry out the assembly according to his own ideas should never be permitted. Undesirable are also designs requiring adjustment or matching of parts during assembly. Assembly errors committed in manufacture can be revealed through quality control. But in actual service, especially if the machine is handled by unskilled personnel, there is no guarantee of its being assembled correctly.

Any uncertainty in assembly involves more labour and time to correct the faults and reduces the efficiency of the assembly. The quality of the assembly in this case depends mainly on the skill of fitters.

An example of a wrong design is shown in Fig. 16*a*. The gear is tightened on the shaft from both ends with two annular nuts *1*. In this design there is no location determining the axial position of the gear and the shaft. Additional time is needed to adjust the posi-

tion of the gear when the unit is built or reassembled. An unskilled or careless fitter is likely to assemble the unit wrongly.

In the design shown in Fig. 16*b* a poor attempt is made to secure the position of the gear. Locating bearing *2* is tightened against the shaft collar *m*. The gear is tightened so that it rests against the inner race of the bearing. If the locating bearing is tightened first and then the gear, the position of the gear is quite definite, but it is also possible that the gear will be tightened first through bearing *3*, and then through bearing *2*. In this case the gear may be displaced from its nominal position.



(a) (b) (c)

Fig. 17. Mounting the blades of an axial compressor

The correct design in Fig. 16*c* has a rigid location in the form of shoulder *n* against which the bearing and the gear are tightened independently. The position of the gear and the shaft is properly secured and may vary only within the machining tolerance limits.

In Fig. 16*d* an overhung gear is mounted in radial-thrust bearings clamped in the housing at both ends with annular nuts. There is no location and the position of the gear in the unit may vary within the stroke of the nuts.

In the correct design shown in Fig. 16*e* the gear is fixed in position by means of a location (bolted-on washer *4*).

The radial position of the blades on the rotor of an axial compressor (Fig. 17*a*) is uncertain. The unit can be assembled correctly only with a special fixture used to adjust the blades to the same distance from the centre of the rotor. In the design in Fig. 17*b* the position of the blades is fixed by a location although it is unilateral. The concentricity of the blades is maintained during assembly by thrusting their bases against the outer cylindrical surface of the rotor. The best designs are those in which the blades are rigidly fixed in both radial directions (Fig. 17*c*).

1.7. Prevention of Wrong Assembly

Not infrequently errors in mounting parts, negligible at first sight and difficult to detect, may derange the operation of the assembled unit and even cause its breakdown. In such cases the correct position of the parts in the assembly should never be indicated by means of marks, notches, etc. The only correct solution is to take proper design measures to ensure the assembly of the parts in the required position only.

In the bearing unit shown in Fig. 18 the cap is located with respect to the housing by two set pins *1* (Fig. 18*a*). The error lies in the symmetrical arrangement of the pins: the cap may be turned through 180° relative to its initial position and then installed; this will impair the cylindricity of the seat and the alignment of the end faces attained during previous machining on the assembled bearing. An asymmetrical arrangement of the pins (Fig. 18*b* and *c*) prevents wrong assembly.

In the sliding contact bearing presented in Fig. 18*d* the shells are installed in a split housing, the upper shell being held by oil-feed sleeve *2* and the lower one, by set pin *3*, both having the same diameter. During assembly the lower shell may be erroneously installed at the top and the upper one at the bottom. The error can be prevented if the sleeve and set pin *3* have different diameters (Fig. 18*e*).

In the bearing unit shown in Fig. 18*f-i* the bush should be installed so that the oil-feed hole in the housing coincides with the hole in the bush. In the design in Fig. 18*f* the bush may be turned by mistake through 180° in which case the oil-feed hole will be shut off.

In the design in Fig. 18*g* wrong assembly is prevented by a check pin *4*. A flat is provided at the oil-hole entrance in the bush to allow for its lower positional accuracy.

This can also be done by means of two diametrically opposite holes with flats made in the bush (Fig. 18*h*).

The bush in the design in Fig. 18*i* is provided with an annular groove that feeds oil with the bush in any position.

Figure 18*j-l* shows cover *5* with a recess connecting two oil holes in the housing. The design in Fig. 18*j* is wrong because the cover may, by mistake, be mounted on the fastening bolts so that the holes in the housing will be shut off. The unit will operate properly if the recess is made in the housing (Fig. 18*k*) and not in the cover, or if the recess in the cover is made cylindrical (Fig. 18*l*).

Figure 19*a-c* shows the installation of a flange with an inner mounting pad *m*. When the fastening bolts are arranged symmetrically (Fig. 19*a*) the pad is likely to be displaced from the required angular position. This can be prevented either by locating the flange with a set pin (Fig. 19*b*) or by placing the fastening bolts in an asymmetric order. The displacement of a single bolt through an angle $\alpha = 5\text{-}10°$ (Fig. 19*c*) will be sufficient to ensure correct assembly.

Figure 19*d-i* shows studs screwed into a housing. In the design shown in Fig. 19*d* the ends of the studs have the same thread, but the lengths of the threaded portions are different, and the studs may be screwed into the housing with their wrong ends.

Various distinctive features such as different shapes of the stud ends, for example, making one end spherical (Fig. 19*e*) or eliminating the smooth portion on the other (Fig. 19*f*) have little effect.

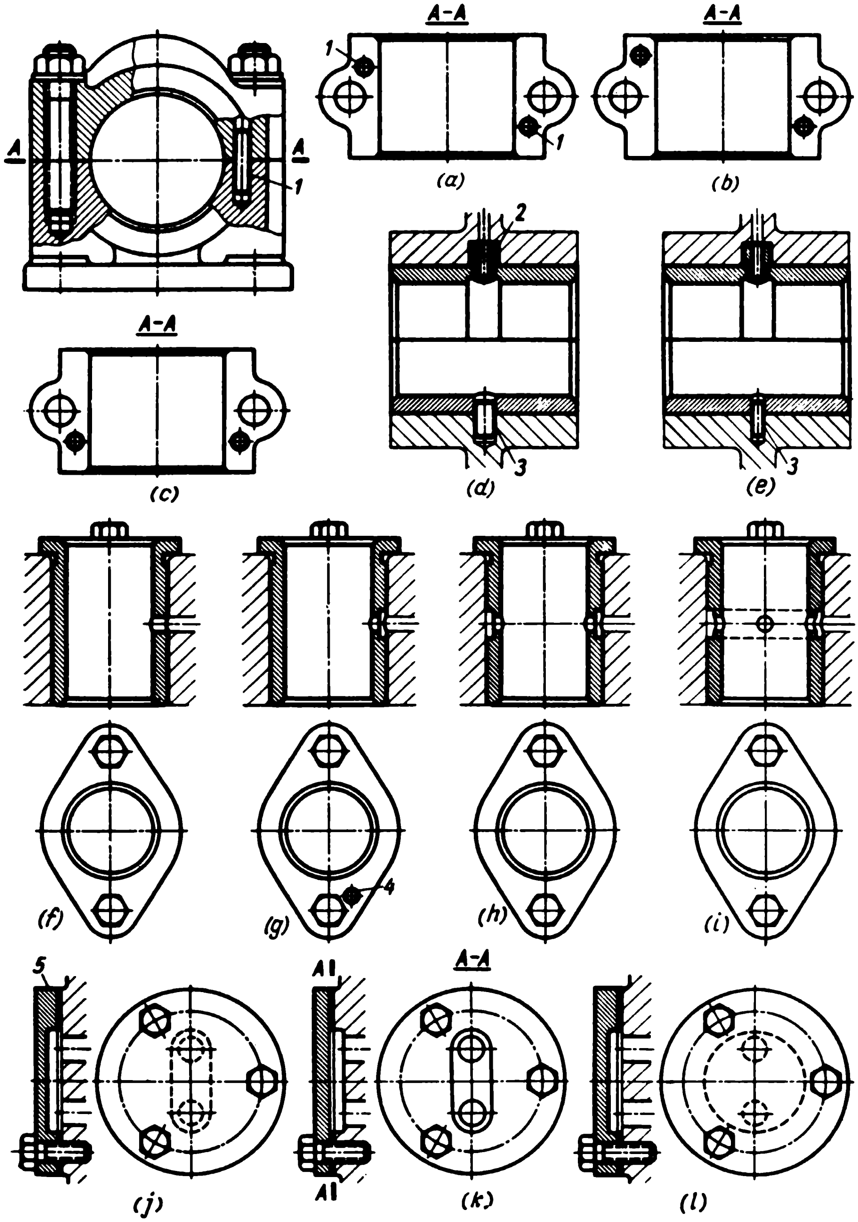


Fig. 18. Prevention of wrong assembly

Assembly will always be correct if the stud threads have different pitch (Fig. 19 *g*) or, better still, different diameters (Fig. 19*h*).

Besides, the ends of a stud may be imparted the same shape and the same axial dimensions (Fig. 19*i*) in which case the position of the stud in assembly is indifferent.

The principle of *foolproof assembly* precludes the possibility of errors, increases the efficiency of assembly operations and saves

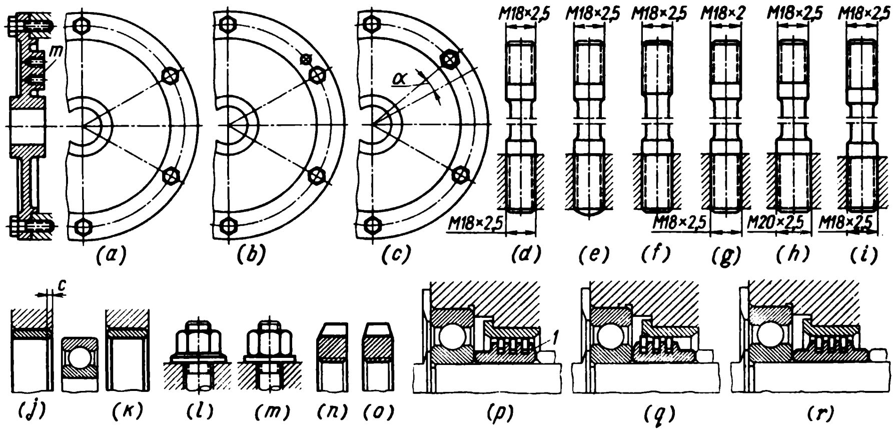


Fig. 19. Prevention of wrong assembly

the fitter's time otherwise necessary to find the proper position of the part.

Figure 19*j* shows a bush press-fitted into a housing. The bush has a slow entry chamfer *c* at one end for installing a rolling-contact bearing. In the case of wrong assembly the chamfer will be on the opposite side making it difficult to install the bearing. In the design shown in Fig. 19*k* where both ends are chamfered the position of the bush during assembly is indifferent.

Fastening nuts with only one chamfer (Fig. 19*l*, *n*) are unpracticable because the fitter must see to it that the nut is placed correctly. In mechanized assembly such nuts delivered to the nut-running tool must be properly oriented. Preference should be given to nuts with chamfers on both sides (Fig. 19*m*, *o*) which can be fitted by either side. Also, it is not advised to employ washers of asymmetric shape (Fig. 19*l*, *m*).

In the oil-seal unit with split spring rings (Fig. 19*p*) seal *1* is asymmetric and must be installed in one position only. The unit will not operate if the installation is wrong (Fig. 19 *q*). In the design in Fig. 19*r* the seal is symmetric and the unit will function properly irrespective of its position.

1.8. Access of Assembly Tools

Fasteners should be easily accessible for assembly tools to facilitate mounting and dismantling. A poor design is illustrated in Fig. 20*a* (mounting of a V-belt drive pulley with a sealing gland). A wrench can approach the bolts of the neck bush only after the pulley is removed from the shaft. In the design shown in Fig. 20*b* the error is corrected by removing the pulley to a distance *s* enough to apply a box-end wrench to the bolt heads.

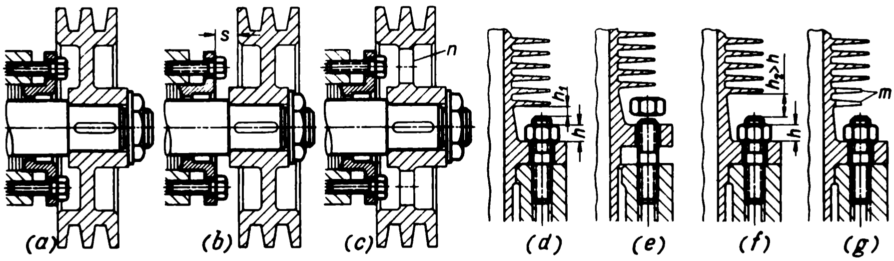


Fig. 20. Access of assembly tools
a, *d*, *e*—wrong; *b*, *c*, *f*, *g*—correct

The disk of the pulley in the design in Fig. 20*c* is provided with holes *n* to admit a socket wrench to tighten up the bolts of the neck bush.

Figure 20*d*-*g* shows the cylinder fastening of an air-cooled engine. The design in Fig. 20*d* is wrong: clearance h_1 between the lower rib and the ends of the clamping studs that remains after the cylinder is fitted onto the studs is less than thickness h of the fastening nuts. This unit can only be assembled by a single, highly inefficient, method: the cylinder is raised up by the studs (Fig. 20*e*) and the nuts fitted on the ends of the studs are then tightened up in succession. For an effective assembly clearance h_2 should be provided between the lower rib and the end of the stud exceeding thickness h of the nut (Fig. 20*f*) or recesses *m* for the nuts made in the lower ribs (Fig. 20 *g*).

Generally, it is recommended that the design should permit the use of *socket wrenches* for screwing nuts and bolts, because these are more convenient to handle, improve the efficiency of the assembly work, damage the nut flats to a lesser degree and enable the tightening force to be increased. In mechanized assembly nuts and bolts are usually screwed with electric or pneumatic tools equipped with socket-type work heads.

Some examples of design changes in fastening units to make them suitable for mechanized assembly are presented in Fig. 21.

In the design in Fig. 21*a* the nuts can only be tightened with an open-ended wrench. Clearance *s* in the design in Fig. 21*b* permits the use of a socket wrench. The most convenient for assembly is the design in Fig. 21*c* where the nuts are arranged on unobstructed surfaces of the part.

In the bracket fastening unit (Fig. 21 *d*) socket-wrench tightening is only possible if the bolts are at a distance *s* (Fig. 21*e*) from the

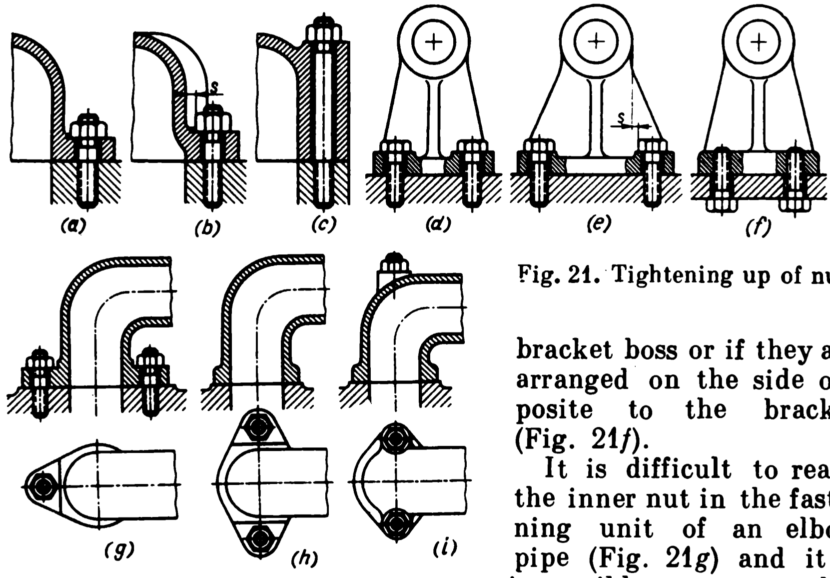


Fig. 21. Tightening up of nuts

bracket boss or if they are arranged on the side opposite to the bracket (Fig. 21*f*).

It is difficult to reach the inner nut in the fastening unit of an elbow pipe (Fig. 21*g*) and it is impossible to use a socket wrench to tighten the nut. In the design in Fig. 21*h* the error is corrected by turning the flange through 90° with respect to the pipe axis. The design in Fig. 21*i* where the nuts are arranged above the pipe surface is still better.

When nuts are arranged in confined places, minimum clearances for wrench application should be assigned that will suit the dimensions of standard nut-runners and their replaceable socket work heads.

The heads of bolts should be locked against rotation during tightening, for example, by butting the hexagon against a shoulder (Fig. 22*a* and *b*) by means of flats (Fig. 22*c*), nibs (Fig. 22*d*), etc., so that the head need not be held by a wrench when the nut is tightened.

It is just as important to prevent the axial displacement of bolts being tightened and prevent them from falling out especially if the assembly is carried out vertically. It is unpracticable to lock the bolts by an annular stop (Fig. 22*e*) since the groove for the stop weakens the bolt. The designs in Fig. 22*f* and *g* are better.

Slow entry chamfers on the ends of fasteners will make nut engagement much simpler when the tightening is done mechanically.

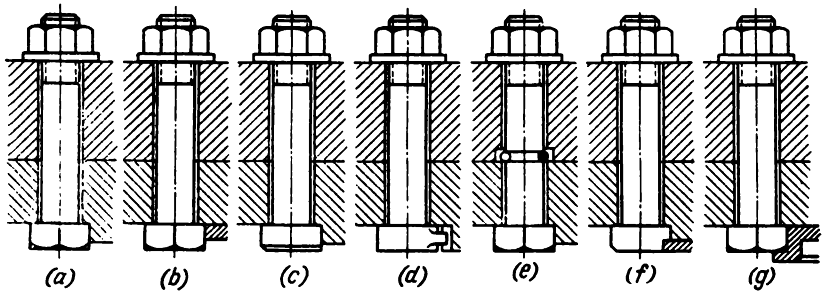


Fig. 22. Locking of bolts against rotation and axial motion

1.9. Rigging Devices

Large, heavy machine components and units should be provided with some rigging devices to enable them to be easily handled during assembly and transportation.

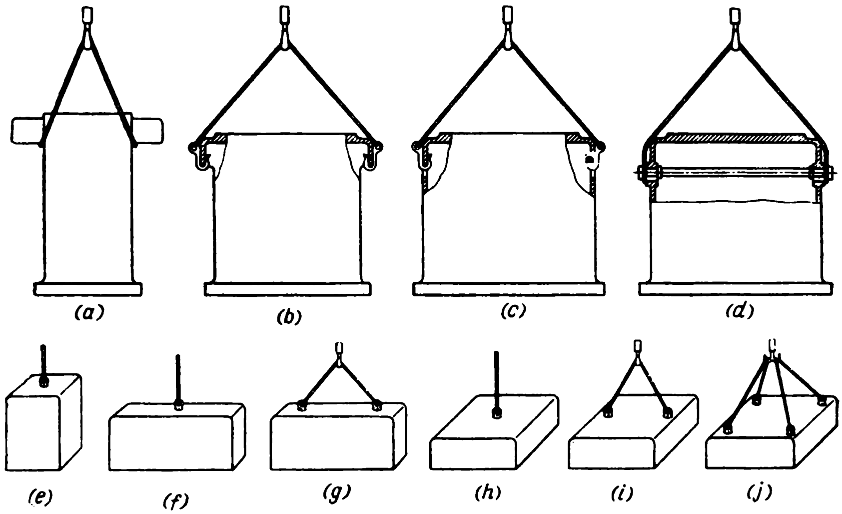


Fig. 23. Suspension of parts in load handling

If the shape of the machine permits, lifting slings and grips are attached to lugs or projections (Fig. 23*a*), flanges (Fig. 23*b*), holes (Fig. 23*c*) or bars passed through the holes (Fig. 23*d*) available on the machine.

If the machine has no such elements it must be equipped with eye-bolts.

A machine or a large part may be suspended from one point only if its centre of gravity is low and the axis of gravity passes through

the suspension point, i.e., when the part is high and has a small cross-section (Fig. 23*e*).

If a wide part is suspended from one point (Fig. 23*f*) it may get out of balance and topple over. Parts of such shape should be suspended from at least two points (Fig. 23*g*). Low, wide and long parts must never be suspended from one or two points (Fig. 23*h*, *i*). Such

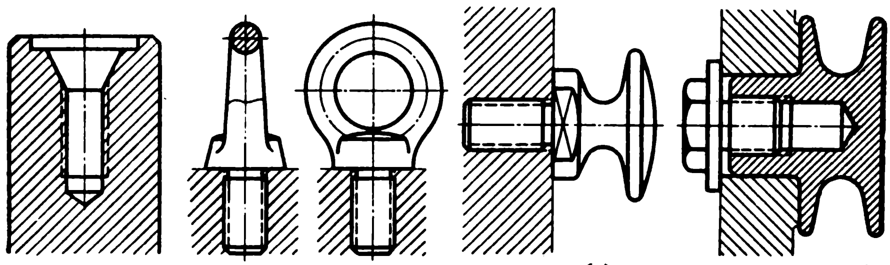


Fig. 24. Rigging bolts

parts should generally be suspended from three or, better still, four points (Fig. 23*j*). Cylindrical shaft-type parts are handled by means of rigging bolts screwed into threaded holes usually located at centre holes (Fig. 24*a*).

Ring bolts (Fig. 24*b*) are most commonly used. Standard ring bolt sizes are selected to suit the acting load.

Cylindrical cantilevered rigging bolts with necks for slings and grips (Fig. 24*c*) are employed for side mounting. Figure 24*d* shows a cantilevered sling bolt intended to carry a heavy load.

Extreme care should be taken when designing non-standard rigging bolts since their poor design may cause a machine to fall from the pulley blocks, thus breaking the machine and possibly injuring personnel. Rigging bolts should have large margins of safety. The use of cast rigging bolts should be avoided. The portions where the bolts contact the lifting slings should be smoothly rounded.

1.10. Spur Gear Drives

During manufacture, the quality of gears is controlled either by checking individual gear elements determining the correctness of engagement (tooth thickness, pitch, runout, tooth profile, etc.), or by testing the gears as a whole against a master gear in a double- or single-profile meshing (without or with backlash, respectively). In the latter case subject to evaluation are the kinematic accuracy of the drive, smoothness of run, backlash, and contact between the teeth. The master gear is made to drive the gear under test, which is being slightly braked, first in one direction and then in the other.

A recorder registers on a profilograph the deviations in the run of the gear under test as compared to that of a calibrated reference gear also meshing with the master gear.

The *kinematic accuracy* is determined by the value ΔF_{Σ} showing the maximum variations in the angular velocity of the gear during one revolution (Fig. 25). This value primarily indicates the run-out of the pitch cylinder with respect to the locating surfaces of the gear (journals, seating, holes).

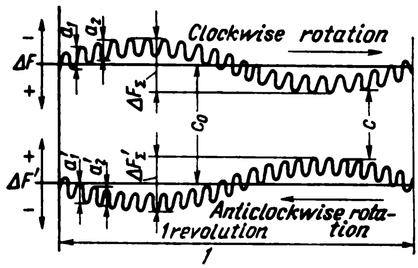


Fig. 25. Engagement chart

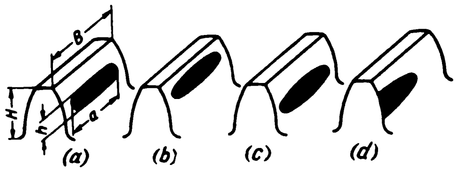


Fig. 26. Contact between teeth

The *smoothness of run* is estimated by the arithmetic mean value of the cyclic errors during one revolution of the gear

$$\Delta F = \frac{a_1 + a_2 + \ldots + a_n}{n}$$

showing the composite error in the tooth thickness pitch and tooth form.

The change in the backlash depending on the angle of the gear rotation is expressed by the distance c between the extreme points of the profilographs for the right- and left-hand rotations, which are separated from each other by distance c_0 equal to the mean backlash value.

Contact between the teeth is checked by applying a thin layer of a marking compound (for example, Prussian blue) onto the teeth of the master gear, rotating the gears and then measuring the gear-contact patterns on the teeth of the gear being tested. Another method consists in coating the teeth of the gear under test with soot and measuring the bright spots on the teeth after rotation.

Tooth contact is characterized by the relative size of the gear-contact patterns (Fig. 26*a*):

over the face width

$$\frac{a}{B} 100\ \%$$

over the depth of tooth

$$\frac{h}{H} 100\ \%$$

where a = mean width of the gear-contact patterns (minus interruptions)
B = face width
h = mean depth of the gear-contact patterns
H = depth of tooth

The displacement of the patterns towards the tooth tip (Fig. 26*b*) shows that the diameter of the pitch cylinder is decreased and their shift to the root of tooth (Fig. 26*c*) shows that the diameter is increased. Contact near the edges (Fig. 26*d*) indicates that the teeth are wedge-shaped or misaligned.

USSR State Standard ГОСТ 1643-56 provides for twelve grades of accuracy in the manufacture of gears (the 1st grade ensuring the lowest and 12th grade, the highest accuracy). Each grade establishes the norms of the kinematic accuracy, smoothness of run, quality of contact, and backlash variations. The choice of the grade of accuracy depends on the purpose of the given gear and the conditions in which it will operate. The kinematic accuracy and smoothness of run are most important for high-speed drives, while the size and arrangement of the gear-contact patterns are of greater consequence for heavily loaded gears. The gears of general-purpose drives are usually manufactured to the 7th or 8th grade of accuracy.

The operating ability of gears in a unit cannot be wholly determined by individual tests of any kind. Apart from the inaccuracies registered by instruments, the operation of a drive is affected by the errors of the centre distances in the housing, inaccuracies in the manufacture of the housing bearings (misalignments) and the faults of the mating gear. Besides, operation under load significantly changes the characteristics of run and contact in view of the elastic deformation of the gear teeth and rims. Heating during operation appreciably changes the amount of backlash.

As a rule, gears heat more during operation than the housing. If the housing is made of cast iron (whose coefficient of linear expansion is about the same as in steel), the heating will reduce backlash. If the housing is manufactured from light alloys whose coefficient of linear expansion is much larger than in steel, backlash can increase.

Example. Calculate backlash in the case of a cast-iron housing ($\alpha = 11 \times 10^{-6}$) and in that of a housing made of an aluminium alloy ($\alpha = 25 \times 10^{-6}$). Given: the working temperature of the gears—100°C and of the housing—50°C. The centre distance is 200 mm.

Heating changes backlash by

$$\Delta c = \Delta A \tan \alpha \tag{1.1}$$

where ΔA = difference between the increase of the centre distance and that of the radii of the gears

α = pressure angle (for a standard gear system $\alpha = 20°$, $\tan \alpha = 0.365$)

For the cast-iron housing

$$\Delta A = 200 \times 11 \times 10^{-6} (50 - 100) = -0.11 \text{ mm}$$

$$\Delta c = -0.365 \times 0.11 = -0.04 \text{ mm}$$

i.e., backlash is appreciably diminished.

For the aluminium housing

$$\Delta A = 200 (25 \times 10^{-6} \times 50 - 11 \times 10^{-6} \times 100) = 0.03 \text{ mm}$$

$$\Delta c = 0.365 \times 0.03 = 0.011 \text{ mm}$$

i.e., backlash is slightly increased.

Possible variations in backlash resulting from the inaccurate centre distance may be found from the relation

$$\Delta' c = \Delta' A \tan \alpha$$

where $\Delta' A$ is the tolerance for the centre distance.

In the case of ordinary accuracy ($\Delta' A = \pm 0.05$ mm)

$$\Delta' c = 0.05 \times 0.365 = 0.018 \text{ mm}$$

Thus, in an unfavourable case (cast-iron housing and the centre distance to the minus tolerance) backlash may become smaller than the nominal value by $0.04 + 0.018 \approx 0.06$ mm.

Except for the thermal ones, most of the other factors affecting the operation of gears are accounted for by checking the backlash between the teeth of the gears mounted in pairs in the housing.

Backlash is commonly checked with a thickness gauge inserted into the spaces between the meshing teeth with the gears in several positions (within one revolution of the gear wheel). With this method, free access to the engagement area must be ensured. If the access is difficult, backlash is determined by swinging one of the gears, with the other being fixed, with the aid of an indicator the contact point of which is applied to one of the accessible teeth in a direction tangential to the pitch circle. Measurements are taken with the gear wheel in several angular positions.

In designs having gears difficult of access backlash is measured by an indicator with a pointer secured to the free end of the gear wheel shaft. Backlash in this case is found by multiplying the measured values by the ratio of the pitch cylinder radius to the arm of measurement.

For a rough check-up, a thin lead strip is passed between the teeth, the thickness of the strip then being measured in sections corresponding to the engagement areas.

The minimum amount of backlash determined by one of the above methods should exceed the possible backlash reduction due to heating, on average, by not less than 0.05 mm.

USSR State Standard ГОСТ 1643-56 establishes backlash values for each grade of accuracy. For medium-accuracy general-purpose drives backlash may be determined from the formula

$$c = (0.04 \text{ to } 0.06)\, m$$

where m is the module.

Contact between the gear teeth is checked with a marking compound. The check-up will only be effective if it is carried out under a load equal to the working load.

The possibilities for adjusting the engagement parameters of spur gears are limited. Should the check-up reveal too small a backlash or unsatisfactory contact, an individual selection of gears is practically the only method of obtaining the needed parameters. This

complicates the assembly. For this reason, when designing gear drives it is important to select such gear accuracy grades, size tolerances, and form of the bearings as would ensure the interchangeability of the gears without complicating excessively the production process.

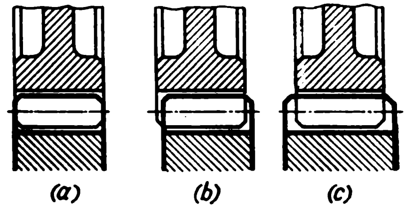


Fig. 27. Mounting of gears

Different hardness is frequently imparted to the teeth of meshing gears to increase their durability and improve running-in. The pinion teeth are hardened, carburized (58-62 Rc) or nitrided (1,000-1,200 VPH) while the gears are structurally improved (30-35 Rc) or medium-temper hardened (40-45 Rc). In such drives the pinions should be made wider than the gears (Fig. 27*c*) so that the pinion teeth overlap the gear teeth whatever are the variations in the axial position of the gears. If the width of the pinions and gears is the same (Fig. 27*a*), a displacement of the gears (because of manufacturing and mounting inaccuracies) will cause a stepwise wear of the softer teeth (Fig. 27*b*) and in the case of subsequent changes in the axial position of the gears this will disturb the engagement.

1.11. Bevel Gear Drives

A frequent error in designing units with bevel gears is that the gears are only fixed in one direction, namely, in the direction of the acting axial forces (Fig. 28*a*), assuming that the gears are fixed in the reverse direction because they thrust against the teeth of the mating gear. Gears should always be fixed in both axial directions (Fig. 28*b*) for the drive to operate reliably and noiselessly, especially under dynamic load conditions.

Provision should be made for the adjustment of the axial position of both gears, for otherwise it will be impossible to match the apices of the pitch cones and obtain the required backlash and satisfactory contact between the working faces of the teeth. The design in Fig. 28*c* is wrong, while that in Fig. 28*d* is correct.

Engagement is usually checked with a marking compound by rotating the drive under a load as near as possible to the working load. Engagement is satisfactory if the gear-contact patterns on all the teeth extend to 0.6-0.8 of the face width and are located in the middle of the tooth (Fig. 29*a*) or closer to the thickened end of the tooth (Fig. 29*b*). The concentration of the gear-contact patterns near the edges of the teeth (Fig. 29*c* and *d*), and especially at the edge of the thinned portion of the tooth (Fig. 29*d*) must not be permitted. The design of a drive should allow for an easy inspection of the gear teeth to dispense with disassembly during each check-up.

The method of adjustment by moving the gears until the end faces of the teeth are matched (on the outer side of the gears) is less accu-

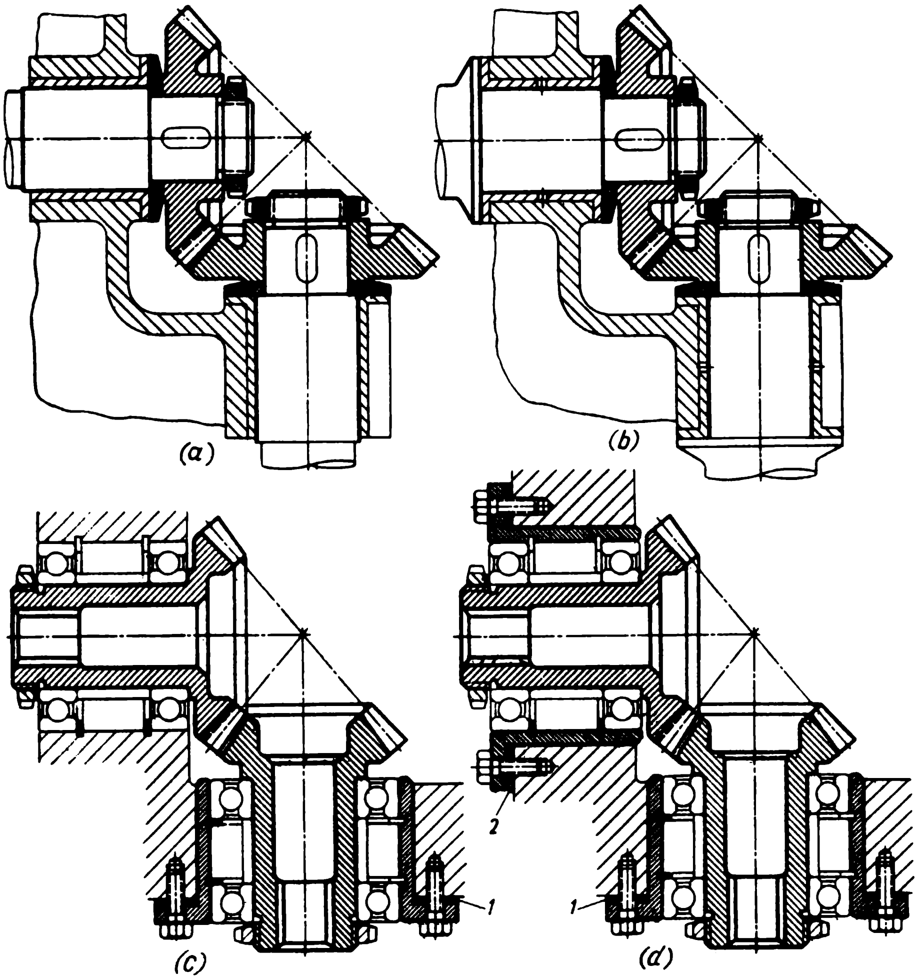


Fig. 28. Adjustment of axial position of bevel gears
1, 2—adjusting washers

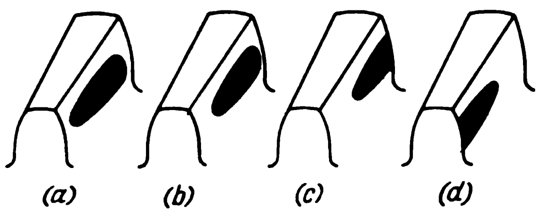


Fig. 29. Location of contact spots

rate. With this method the end faces of the teeth in the engagement area should be open to view.

Because of the smaller accuracy in the manufacture of bevel gears the backlash in such gears is made slightly greater [(0.06 to 0.1)m, where m is the module]. The backlash between meshing gears is checked either with a thickness gauge introduced into the spaces between the meshing teeth from their ends (on the outer side of the gears) or with an indicator the contact point of which is applied to one of the teeth or to a pointer secured on the gear shaft.

There are two methods of adjusting the axial position of gears.

With the first method the position of the *gear on the shaft* is changed. The shaft secured by its bearing surfaces remains in place. This method can be only applied if the gear is not made integral with the shaft.

With the second method the *gear is shifted together with the shaft.* This method may be applied if the change in the axial position of the shaft within the adjustment range (usually 0.5-1mm) does not affect the operation of the parts mated with the shaft.

Otherwise it is necessary to divide the shaft into two portions, one of which can be shifted axially while the other is fixed in the axial direction, and connect both portions by means of a compensator (for example, a splined compensator).

This is the only possible method for gears made integral with the shaft. It is also frequently employed for fitted-on gears.

Some methods of adjusting the axial position of gears mounted in rolling contact bearings are illustrated in Fig. 30.

The axial position of a gear *on a shaft* is commonly adjusted by means of changeable calibrated washers *1* (Fig. 30*a*). For adjustment the gear has to be taken off the shaft in which case the unit must be disassembled. In order to make the adjustment easier, the calibrated washers are manufactured in the form of half-rings *2* (Fig. 30*b*) inserted into a recess made in the gear. In this case it is enough to shift the gear on the shaft to a distance equal to the depth of the recess after which the half-rings can easily be removed and replaced by other ones.

The gear can be made to shift together with the shaft by replacing thrust washers *3* (Fig. 30*c* — gear made integral with the shaft; Fig. 30*d* — fitted-on gear).

Figure 30*e-j* shows the adjustment by shifting the bearing housing.

In the design in Fig. 30*e* the adjustment is done by means of a set of shims *4* made of metal foil and placed under the housing flange. The shortcoming of the method is that the unit has to be disassembled.

In the design shown in Fig. 30*f* the adjustment is done by replacing calibrated half-rings *5* fitted into a recess made in the housing flange. It is enough to move the housing forward to a distance equal to the depth of the recess to replace the half-rings.

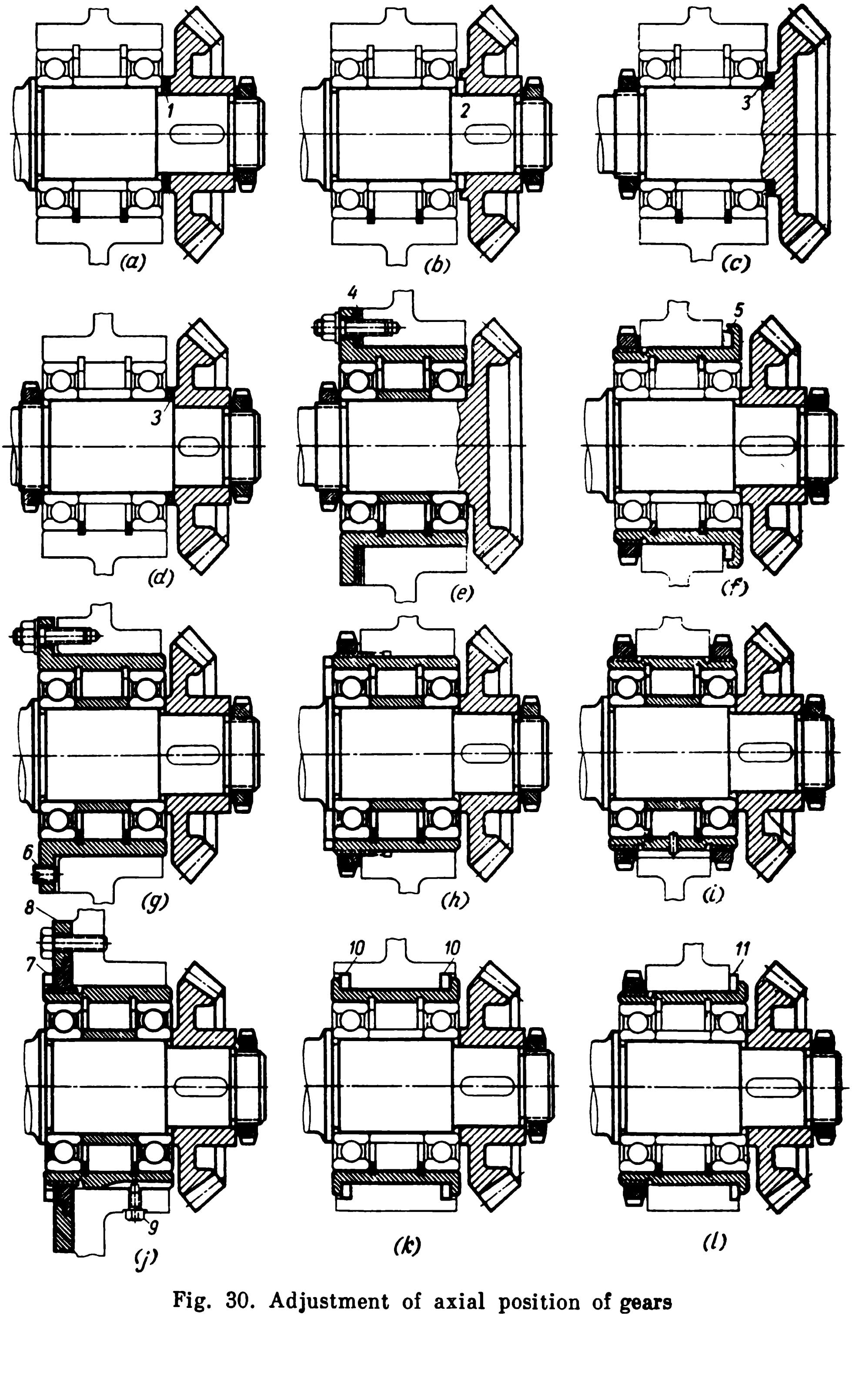


Fig. 30. Adjustment of axial position of gears

In the design in Fig. 30*g* the adjustment is carried out without disassembling the joint with the aid of pressure screws *6* (usually three in number). In order to move the gear towards the centre of the drive it is necessary to slacken the screws by the required amount and then tighten up the fastening bolts. To move the gear away from the centre of the drive one should unscrew the fastening bolts

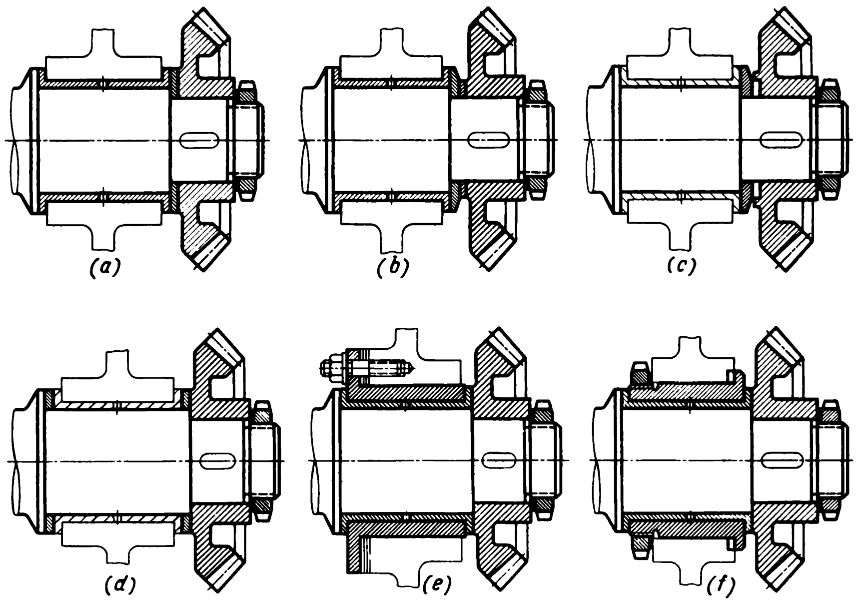


Fig. 31. Adjustment of axial position of gears

and then screw in the adjusting screws. An essential shortcoming of this design is that it is difficult to locate the housing from three points simultaneously; as a result, there is a possibility of the housing being skewed when tightening the bolts.

In the design shown in Fig. 30*h* the axial shift is effected by turning the housing which is thread-fitted in the bed (with a smooth centring portion). The adjusted housing is secured by means of a lock nut.

In the design in Fig. 30*i* the housing is shifted in the axial direction by means of annular nuts installed on both sides of the housing.

All these methods slightly impair the centring of the shaft because the housing must be installed by a slide fit.

Convenient adjustment is shown in Fig. 30*j*. Here, the housing is shifted by rotating annular nut *7* screwed onto the housing and fixed in the axial direction by washer *8*. The rotation of the bearing

housing is prevented by screw *9*. The joint will inevitably have an end play equal to the sum of the clearances in the thread and on the faces of the annular nut. As distinct from the designs shown in Fig. 30*e-i*, the housing is not tightened, an undesirable feature under dynamic loads.

In the design in Fig. 30*k* (radial assembly) the adjustment is done with the aid of half-rings *10* (a loose joint), and in Fig. 30*l*, by means of half-rings *11* tightened with a nut.

The methods of adjustment for gears mounted in sliding contact bearings are presented in Fig. 31. In the designs in Fig. 31*a-c* the gear is shifted along the shaft and in the designs in Fig. 31*d-f*, together with the shaft.

1.12. Spur-and-Bevel Gear Drives

As distinct from bevel gear drives in which the generatrices of the active surfaces of the teeth converge at the point of intersection of the gear axes (Fig. 32*a*), in the combined *spur-and-bevel gear drives* one of the gears (pinion) has straight teeth (Fig. 32*b*). In the mating gear the tooth spaces correspond to the pinion tooth profiles, i.e., the generatrices of the spaces are mutually parallel and the teeth become thinner towards the centre of the gear to a larger degree than in ordinary bevel gears.

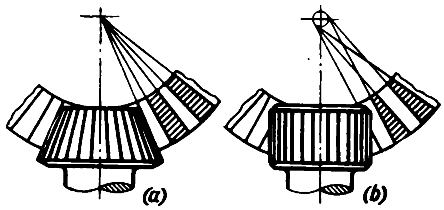


Fig. 32. Gearing diagrams
a—bevel drive; *b*—spur-and-bevel drive

Friction without sliding over the tooth width usually observed in ordinary bevel gear drives is absent in the spur-and-bevel gear drives. In many cases this fact is immaterial. Pure rolling friction in any involute gearing is only observed in tooth sections close to the pitch circle; sliding friction is added to rolling friction at the root and the top of the tooth. Sliding over the tooth width also occurs in drives with skew gear axes, but nevertheless this does not prevent these drives from operating reliably for a long time.

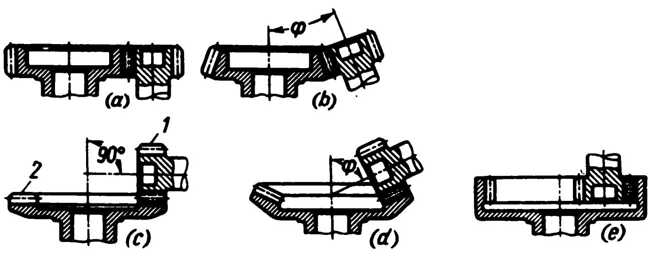


Fig. 33. Spur-and-bevel drives

In the combined spur-and-bevel gear drives sliding diminishes as the angle φ between the gear axes becomes smaller (Fig. 33*b*-*d*). When $\varphi = 0$ (Fig. 33*a* and *e*) a spur-and-bevel gear drive becomes a purely spur gear drive. Sliding is reduced with a smaller face width with respect to the diameter of the gear, and with a higher gear ratio.

Spur-and-bevel gear drives are manufactured on the gear-cutting machines used for spur gears. Pinions *1* with straight teeth are machined by the usual shaping and milling methods, and the mating gears with wedge-shaped teeth are generated using a gear cutter whose shape corresponds to that of the spur pinion. Both can easily be ground, and a high surface hardness can be imparted to their teeth.

Helical teeth are cut by the usual shaping methods with a helical gear cutter.

The spur pinion (with straight teeth) is not subjected to axial pressure and does not require any axial adjustment if its teeth overlap those of the bevel gear.

Spur-and-bevel gear drives can be engaged and disengaged by moving the spur pinion, in the same way as the ordinary spur gear drives.

Spur-and-bevel gear drives are employed with small and medium torques and with gear ratios from 1 and higher. Such drives are known to be used in high-power installations.

Convenience in Maintenance and Operation

When designing units, assemblies and machines one should provide for their convenient maintenance, operation, disassembly, reassembly and adjustment, make them easily accessible for inspection, and prevent their possible breakdowns due to unskilled or careless handling.

Also, the machine should have an attractive external appearance.

2.1. Facilitating Assembly and Disassembly

Let us consider some examples of how to facilitate the assembly and disassembly of connections which have to be frequently dismantled when in use (Fig. 34).

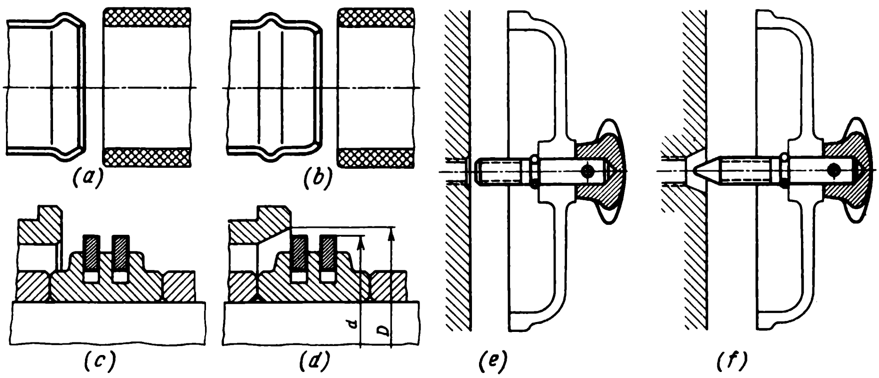


Fig. 34. Ways to facilitate assembly

It is difficult to fit a flexible hose onto the pipe shown in Fig. 34*a*. In the design in Fig. 34*b* the guiding portion with rounded-off edges makes the process much more easier.

In seals with split spring rings (Fig. 34*c*) the assembly is simplified if the housing is provided with a slow entry chamfer of diameter D exceeding the diameter d of the rings in their free state (Fig. 34*d*).

In the case of hard-to-reach joints, especially with blind assembly, it is good practice to provide the male parts (Fig. 34*e*) with a taper, and the holes, with guiding cones (Fig. 34*f*).

The inner spaces and ducts of oil systems should periodically be cleaned to remove dirt and the products of thermal decomposition of oil. Oil ducts should preferably be plugged up (Fig. 35*c*, *d*) and not sealed permanently as shown in Fig. 35*a*, *b*.

Figure 36*a* illustrates an irrational design of the oil space in the neck of a crankshaft. The space is sealed by end caps made of sheet steel and press-fitted into the crankshaft webs. The space can be

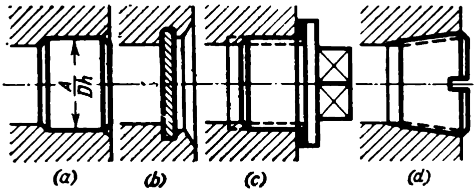


Fig. 35. Sealing of oil ducts

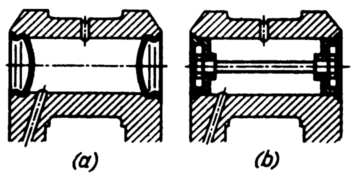


Fig. 36. Sealing of oil spaces in a crankshaft

cleaned only by injecting a washing solution into the interiors of the shaft. The design with detachable caps (Fig. 36*b*) is far more better.

Joints which are frequently disassembled and assembled when in use should be made readily detachable. Figure 37 shows the tip of an ignition system conductor. In the design in Fig. 37*a* the fastening nut of the contact screw has to be unscrewed completely to remove the conductor. In the design shown in Fig. 37*b* where the conductor has a split tip it will be enough to unscrew the nut to the height h of the fixing flange on the tip to remove the latter from the screw.

Figure 38 illustrates a quick-acting clamp with a swing bolt (frequently used to fasten the covers of autoclaves). The nut is unscrewed to the height ensuring its free passage over the corner of the cover and the bolts are then swung back to release the cover.

The fastening of a cylindrical part in a spring U-clamp is illustrated in Fig. 39.

Quick-acting connections widely employ clamps with a swing arm. The clamp operating on the toggle principle consists of arm *1* (Fig. 40*a*) swinging on pin *2*. Stirrup *3* engaging the hook of part *4* being tightened is attached to the arm. When the arm is swung to the position shown in Fig. 40*b* it tensions the hook. By the well-known property of the toggle mechanism the tension reaches its maximum at the dead centre. Beyond the dead centre (angle α) the

arm is secured by the elastic forces of the system which press the arm against stop *m*.

The use of such a snap-action toggle for fastening a cylindrical tubular part *1* is exemplified in Fig. 41.

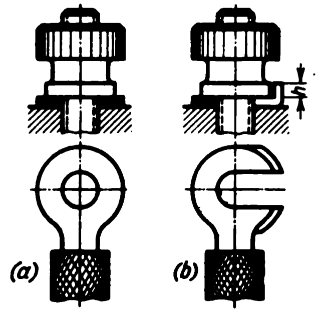


Fig. 37. Tip of a conductor

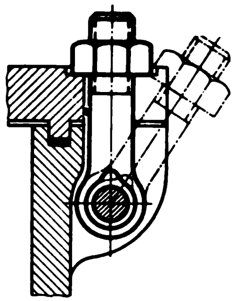


Fig. 38. Swing bolt

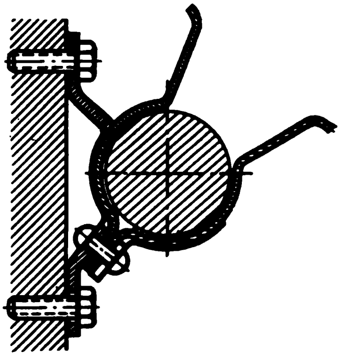


Fig. 39. Spring clamp

Figure 42 illustrates the adjustment of the axial position of a shaft in a split sliding-contact bearing by means of adjusting rings (radial assembly). In the design in Fig. 42*a* the adjusting rings *1* are solid. To carry out the adjustment, it is necessary to take off bearing cap *2*, remove the shaft and take off the fitted-on part *3*.

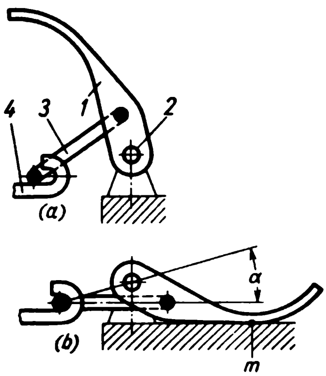


Fig. 40. Fast-acting lock

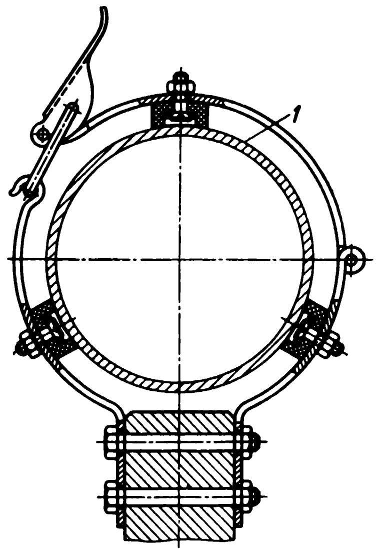


Fig. 41. Tubular part fixed by fast-acting lock

In the design shown in Fig. 42*b* where the adjusting rings are split (half-rings *4*, *5*) it is only necessary to remove bearing cap *2*

and then, leaving the shaft in place, remove half-rings *4* and then, half-rings *5* after turning them around the shaft axis through 180°.

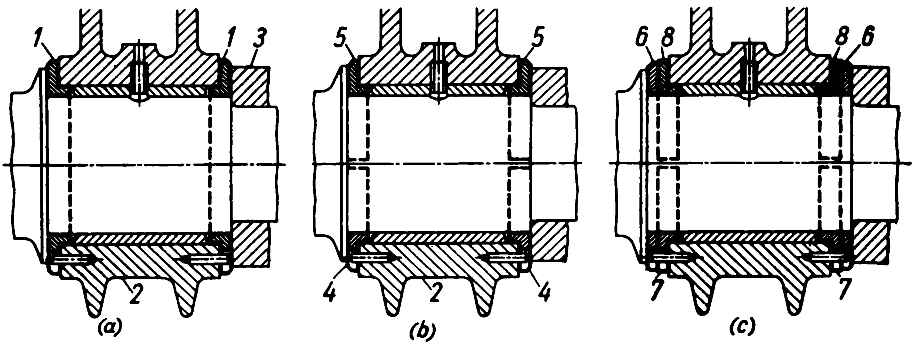


Fig. 42. Adjustment of axial position of a shaft

If the operating conditions require a full bearing surface not interrupted by splits, additional solid rings *6* are introduced (Fig. 42*c*).

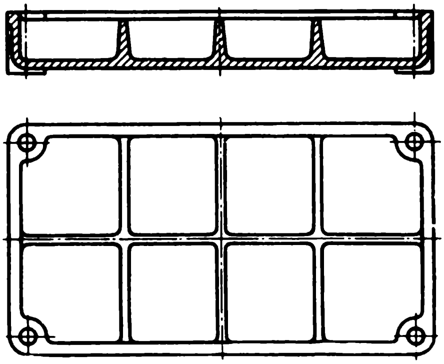


Fig. 43. Cover with compartments for fasteners

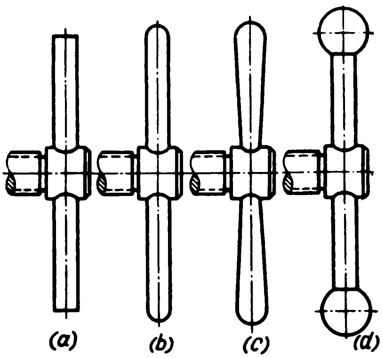


Fig. 44. Designs of jack screw handles

During adjustment these rings remain in place. Adjusting half-rings *7*, *8* can be taken off without disassembling the shaft.

For a more convenient disassembly and reassembly the detachable covers of housing-type components should be provided with partitions (Fig. 43) to form several compartments for the taken-off fasteners, each compartment accommodating fasteners of a definite size and type.

Handles, handwheels, hand nuts, etc., should have convenient shape.

Figure 44*a* presents an irrational design of a jack screw handle. Improved designs are illustrated in Fig. 44*b*, *c*, *d*.

Knurled nuts (Fig. 45*a*) cannot be tightened forcibly by hand. In the design in Fig. 45*b* the sharp edges of the nut may cut the fingers. Besides, a dirt trap is formed in the upper hollow of the nut.

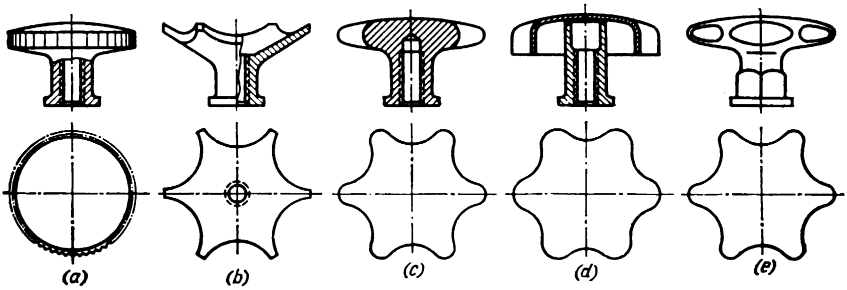


Fig. 45. Hand-driven nuts

Correct designs that permit good tightening by hand are illustrated in Fig. 45*c* and *d*. If hand nuts are to be forcefully tightened, use is made of additional elements in the form of flats or hexagons (Fig. 45*e*).

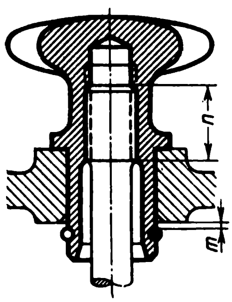


Fig. 46. Attached nut

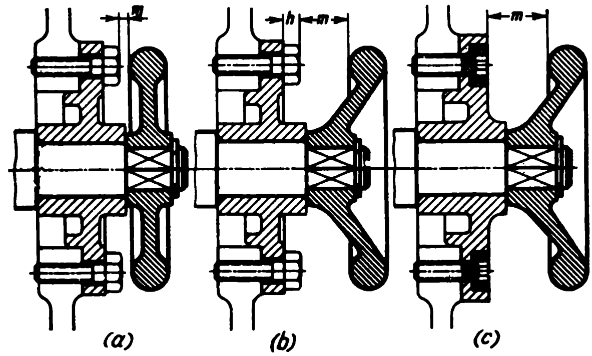


Fig. 47. Designs of handwheels

To accelerate and simplify the assembly of joints which are frequently disassembled while in use it is good practice to employ the so-called non-losable nuts which are held in the part being attached, for example, by means of circlips (Fig. 46). Each single nut is held with a minimum axial clearance *m*. Such nuts are used as pullers. In a joint with several nuts the axial clearance *m* should slightly exceed the length *n* of the bolt thread. Otherwise, it will be difficult to screw the nuts on and off (as all nuts would have to be turned in succession by a small amount each time to avoid the misalignment and pinching of the part).

Hand nuts, handwheels, etc., should be designed to provide free access to the hand and a firm grip. The handwheel design shown in

Fig. 47*a* is wrong. The small clearance *m* between the handwheel and the fastening bolts does not admit the hand. In the design in Fig. 47*b* the handwheel rim is farther from the housing wall. If sunk hexagonal bolts are used (Fig. 47*c*) the clearance *m* is increased by the height of the bolt head *h*.

The minimum clearance *m* necessary to conveniently grip a handwheel is equal to 20-25 mm. It should never be less than 35-40 mm for machines operating in open air, especially if gloves are worn.

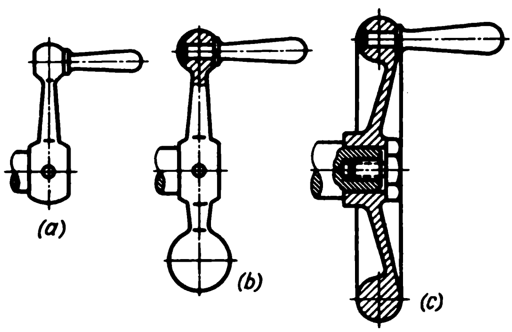


Fig. 48. Designs of handles

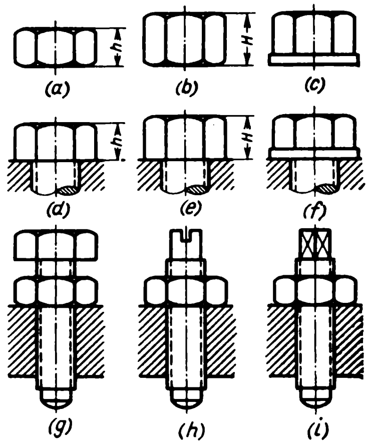


Fig. 49. Designs of nuts and bolts

Handwheels and handles intended for rapid rotation (for example, the gear shifting handles of metal-working machines, handwheels used in worm drives, etc.) should possess an increased flywheel mass which makes it easier to overcome the nonuniform torque of the drive. Drive handles (Fig. 48*a*) should carry counterweights (Fig. 48*b*) or be made in the form of handwheels with massive rims (Fig. 48*c*).

Hand-operated controls should be polished to the 11th or 12th class of surface finish to prevent injury to the hands, improve external appearance and avoid corrosion.

For general-purpose screw joints one should use nuts chamfered on both faces, which may be installed by either side.

For screw joints which have to be frequently taken apart while in use, it is advisable to apply thicker nuts and bolts with taller heads [$H = (1 \text{ to } 1.4)\, d$], as in Fig. 49*b*, *e*, instead of ordinary nuts and bolts [$h = (0.7 \text{ to } 0.8)\, d$], as in Fig. 49*a*, *d*, and increase their hardness (35-40 Rc) in order to prevent the crushing of their flats.

A collar at the base of the hexagon (Fig. 49*c*, *f*) makes the nut application easier for it prevents wrench slip. However, this design

is not suitable for mass production (as the nuts cannot be manufactured from hexagonal rolled stock).

Whenever the design permits, box or socket wrenches should be used.

As a rule, the hexagon dimensions should be unified as much as possible to reduce the size range of wrenches. But if the bolts are

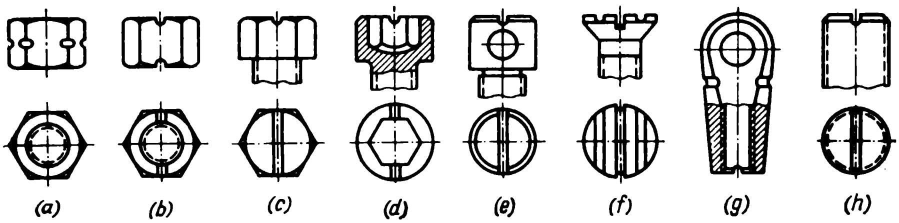


Fig. 50. Distinction marks for fasteners with left-hand thread

secured by lock nuts, it is advisable to use different tools for the bolts and the nuts (Fig. 49*h*, *i*). In the case of identical hexagons (Fig. 49*g*) one has to keep duplicate wrenches in his tool set.

Nuts and bolts with left-hand threads should be marked to prevent unscrewing them in the wrong direction, as this may cause

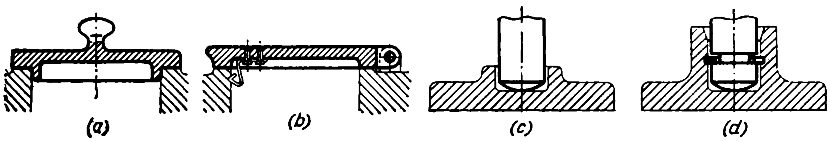


Fig. 51. Structural attachment of parts
a, *c*—wrong; *b*, *d*—correct

damage to the clamped parts. Such marks for fasteners with left-hand threads are illustrated in Fig. 50*a-h*.

Individual parts belonging to the basic outfit of a machine should be structurally attached to it as loose parts may be lost when the machine is transported or repositioned. Examples are shown in Fig. 51*a*, *b* (an inspection cover) and *c*, *d* (a leg with a self-aligning shoe).

2.2. Protection Against Damage

Measures should be taken to safeguard the brittle elements of machine components and their precision surfaces against careless handling.

Let us take, by way of example, the head of an air-cooled engine cylinder made of an aluminium alloy (Fig. 52*a*). The thin ribs can

be safeguarded against breakage by making the lower rib thicker (Fig. 52*b*) or press-fitting a steel rib on the cylinder (Fig. 52*c*).

The end faces of splines will be effectively protected from dents, in the case of chance impacts, dropping, etc., by chamfers of diameter D that exceed the major diameter D_0 of the splines (Fig. 52*d*)

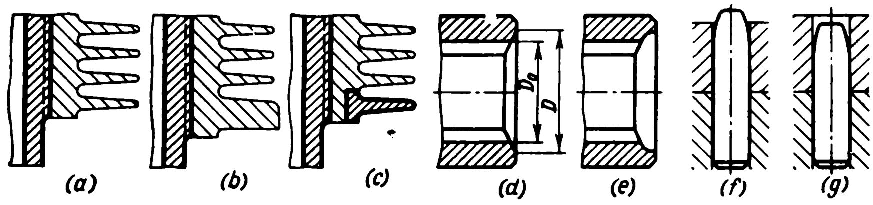


Fig. 52. Protection of design elements against damage

or by sinking the splines with respect to the end face of the part (Fig. 52*e*).

To avoid damage, set pins (Fig. 52*f*) should be sunk in the part being located (Fig. 52*g*).

Parts carrying the heaviest stress are liable to fail and for this reason measures should be taken to prevent their breakdown with the resulting serious damage to the machine.

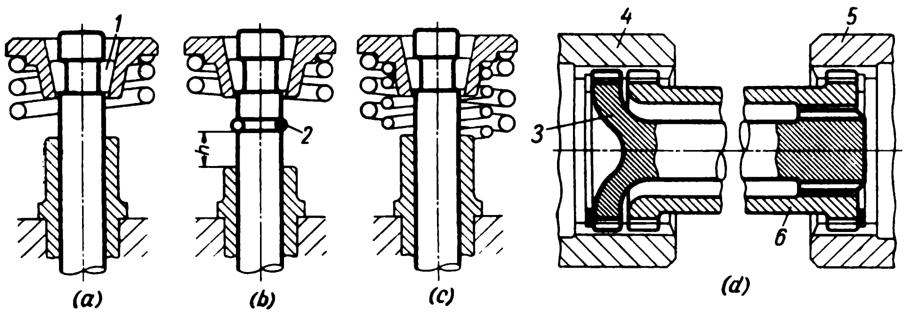


Fig. 53. Protection against the consequences of breakdown

One example is the valve of an internal-combustion engine (Fig. 53*a*). Should the valve spring break, the valve hangs in the guide and hits the piston crown, and if, in addition, taper valve retainer blocks *1* leave their seats the valve drops into the cylinder. The result is a serious breakdown because of the valve stem butting against the combustion head.

In the design shown in Fig. 53*b*, the breakdown is prevented by retainer ring *2* fixed on the stem at a distance h from the end face of the guide, the distance somewhat exceeding the valve stroke.

Two (Fig. 53*c*) or three concentric valve springs will practically exclude the possibility for the valve to fall into the cylinder. The coils of the adjacent springs are oppositely inclined so that if one spring breaks, its coils do not get into the spaces between the coils of the adjacent intact spring.

Figure 53*d* shows torsion spring *3* used for an elastic transmission of torque from shaft *4* to shaft *5*. As in all other springs, higher design stresses are adopted for torsion springs and, as a result, their failure in the case of overloads, for example, when torsional oscillations develop, cannot be excluded.

To prevent overloads, the spring is enclosed in splined sleeve *6* meshing with the same splines as the spring but with a larger backlash. The torsion spring operates in its normal conditions. When the rated torque is exceeded the load is taken up by the sleeve, which prevents the spring failure. If the spring breaks the torque is transmitted by the sleeve, although with reduced elasticity.

2.3. Interlocking Devices

Machines and their units should be reliably protected against damage that may be caused by careless or clumsy handling. The machine must be designed so as to exclude any possibility for its wrong operation. In machine tools this is achieved by means of

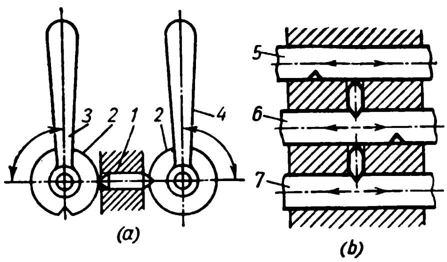


Fig. 54. Interlocking devices

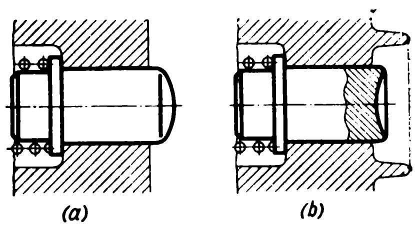


Fig. 55. Prevention of an accidental switching of push buttons
a—wrong; *b*—correct

automatic interlocking devices which cut out the machine or its mechanisms in the case of overtravels. In change-over mechanisms provision should also be made for devices that will not allow simultaneous engagement.

Figure 54*a* shows the hand drive of directional control valves. The conditions of operation require that each valve be turned only when the other one is in a definite position. This is done by means of lock pin *1* controlled by disks *2* rigidly attached to the actuating handles. When handle *3* is turned, handle *4* is held in place by the

lock pin. Handle *4* can only be turned when handle *3* is in a definite position.

An interlocking device widely applied in gearboxes in which the gears are shifted by means of selector bars is illustrated in Fig. 54*b*. Bar *5* can only be moved when bars *6* and *7* are locked, bar *6* can be shifted when bars *5* and *7* are locked, and bar *7*, when bars *5* and *6* are locked. Thus, this device allows for each gear to be engaged only when all the other gears are brought out of mesh.

The problem can often be solved by introducing mechanical links between the elements to be shifted, the drive being effected in a centralized manner by means of a single handle (*single-handle control*).

The design of hand-operated push buttons should be such as to prevent accidental switchings. Protruding push buttons (Fig. 55*a*) are not safe because they may be accidentally depressed. Sunk buttons (Fig. 55*b*) are the best design.

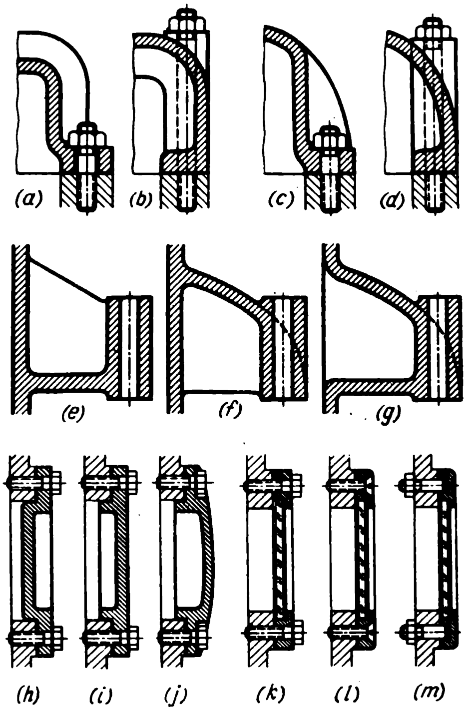


Fig. 56. Elimination of recesses and protruding parts

2.4. External Appearance and Finish of Machines

The machine as a whole and its structural elements must have smooth outlines. This is a very important provision for facilitating the maintenance of the machine and keeping it tidy.

Undesirable are high ribs, sharp corners and cavities which accumulate moisture, dirt and dust, making it difficult to wipe and wash the machine. It is more practicable to replace outer ribs (Fig. 56*a*) by inner ones (Fig. 56*b*). Fasteners should never be arranged in recesses (Fig. 56*c*). It is better to place them beyond the surface of the fastened part (Fig. 56*d*).

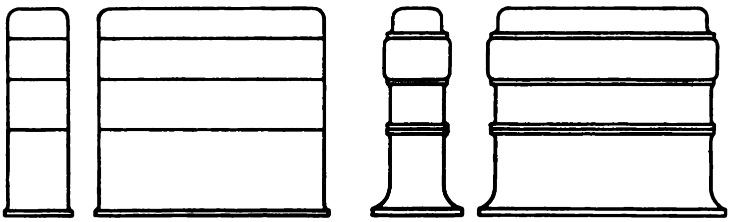
Figure 56*e* shows a poor design of a trough-shaped lug. It is difficult to clean the trough of dirt accumulating between the ribs. A better design closed on the top is presented in Fig. 56*f*, but the closed box-shaped design shown in Fig. 56*g* is the best.

Recessed covers (Fig. 56*h*) should be avoided. Flat (Fig. 56*i*) or slightly convex (Fig. 56*j*) covers are more preferable.

In the sight glass fastening (Fig. 56*k*) the protruding heads of the bolts impair the general appearance and make it difficult to wipe the glass clean. The design in Fig. 56*l* is better because the bolts are replaced by countersunk screws. In the best design shown in Fig. 56*m* the outer surface is smooth and the glass frame is fastened from the internal side of the housing by means of studs resistance-welded to the frame.

Aesthetic aspects are as important. Smooth, streamlined contours are undoubtedly pleasant to the eye.

The aesthetic aspect of a machine in the first place is determined by its engineering reasonableness. When a rational compact layout



(a) (b)

Fig. 57. Machine housing shapes
a—irrational, *b*—rational

is combined with an effective power scheme the machine always have a beautiful appearance. A machine with slap-dash units, with open operating members, with openings and hollows between the structural elements loses much in its appearance.

For all their compact layout and smooth external appearance machines should never take the form of mere box-like structures. It is expedient to adhere to a definite architectural pattern agreeing with the shape of the machine and accentuating its general horizontal or vertical design. Such a pattern can be produced by using cornices, ribs, abutting welts, etc., emphasizing in relief the principal structural elements.

A machine with a box-like shape and having smooth joints (Fig. 57*a*) produces an impression of a heavy block of metal. The machine assumes a much lighter and well-proportioned appearance if the alternating horizontal components are made of slightly different length and width with welts along the contour of the joints (Fig. 57*b*).

The welts have not only a decorative, but also a practical purpose. They can be filed to correct casting inaccuracies and match the contours of the contacting surfaces.

It is advisable to enliven long surfaces, panels and shields by a simple and austere reliefed pattern that conforms to the shape of the machine, for example, in the form of parallel ribs directed horizontally or vertically, depending on the general design of the machine. Besides, reliefs increase the rigidity of the shields.

Much attention should be given to the arrangement, external appearance and finish of control members. They should be mounted near the operator's station, in a place convenient for manipulation and inspection, and, as far as possible, on a single panel. It is good practice to polish metal parts or coat them with chromium or coloured enamels. Glittering coatings (decorative chrome plating) should be avoided because they fatigue and even blind the eyes with bright illumination. Lustreless chrome plating is most effective.

All sorts of trade marks, tables indicating the parameters, diagrams, etc., should be imprinted on massive plates in clear and large characters by means of phototype or engraving (but not punched on thin tin sheets). The plates should be positioned in a place convenient for reading, and, if necessary, illuminated (if installed in recesses or boxes).

A beautiful finish of a machine will without any doubt make the personnel treat it with greater care.

A machine should never be excessively beautified. Abundant glittering surfaces, diversity of colours, bright and flashy hues in the finish will impair the external appearance of the machine. The finish of a machine should be technically justified, correspond to the functional purpose of the parts and make control and servicing easy and convenient. The forms should be simple and austere, and the colours, serene.

It is good practice to paint machines operating in enclosed premises with light colours (pale blue, light green, light grey) which possess a higher reflection coefficient and intensify the illumination of the premises. Where sanitation is the prime demand (food industry, medicine) preference should be given to milky-white or ivory colours.

Machines operating in open air and subjected to the action of dust, soot, exhaust gases, etc., should preferably be painted with dark colours.

A coating should be durable and wear resistant, proof against atmospheric effects, possess good adhesion to metal surfaces and reliably protect the metal against corrosion. Oil varnishes are now being ousted by new, more stable synthetic coatings (nitrocellulose enamel, escapone varnishes, alkide, phenolic and epoxy coatings, etc.). Organo-silicon coatings are the best. They effectively repel water, dust and dirt and are stable against light and heat.

Designing Cast Members

Casting is widely used for making shaped parts ranging from small elements to very large beds and housings. In many machines (internal-combustion engines, turbines, compressors, metal-cutting machine-tools, etc.) the weight of cast parts comes to 60-80 per cent of the total machine weight.

Casting can produce most intricately shaped parts which cannot be made by any other forming method. The casting process is highly productive and inexpensive.

Characteristic of cast parts are reduced strength, differences in mechanical parameters between their different portions and liability to the formation of internal defects and stresses. The quality of a casting depends on its design as well as manufacturing process. For this reason the designer must know the basic casting practices and the methods for obtaining high-quality castings at the minimum production costs.

The following casting methods are commonly in use.

Sand mould casting. This is the most widespread and universal method and practically the only one used to make large-size castings. Moulding is done to wooden or metal patterns in flasks packed with sand-clay mixtures. The internal cavities in castings are formed by means of *cores* moulded from sand mixtures with binders in core boxes.

The dimensional accuracy of a casting depends on the quality of the mould manufacture and properties of the casting alloy (average deviation from nominal dimensions is $\pm 7^0/_{00}$). The surface finish is within class 3-4.

The efficiency of the casting process and the quality of castings are appreciably improved by using *mehcanical moulding* when the flasks are packed with squeeze moulding, jolt moulding and sand-throwing machines.

Critical and large-size parts are cast in *core moulds* the external and internal surfaces of which are formed by blocks of cores connected mechanically or by bonding.

Shell mould casting. The moulds in the form of shells 6-15 mm thick are prepared to metal patterns from a mixture of sand with a thermosetting resin (bakelite) which is then set by heating to 150-350°C. This method is mainly employed to cast open (through- or cup-shaped) parts with a size of up to one metre. The dimensional accuracy is $\pm 5^0/_{00}$ and the surface finish, up to the 6th class.

Chill casting. Metal is poured into permanent iron or steel moulds (chills). For small-size and nonferrous castings the internal cavities are formed by metal

cores and in the case of medium- and large-size castings, by sand cores (*semi-permanent mould casting*). This method provides for increased strength of the castings, an accuracy of $\pm 4^0/_{00}$ and surface finish of up to the 6th class.

Centrifugal casting. This method is utilized to cast hollow cylindrical components such as pipes. Metal is poured into revolving cast-iron or steel drum-shaped moulds where it is compacted by centrifugal forces. The casting accuracy (wall thickness) depends on the accuracy of metering the metal feed.

Small parts are cast on centrifugal machines with permanent metal moulds.

Investment casting. Patterns are made of easily fusible materials (paraffine, stearine, wax, colophony) by pressure casting into metal moulding dies. The patterns are joined into blocks, coated with a thin layer of a refractory material (quartz powder with ethylsilicate or liquid glass) and moulded into unsplit sand moulds which are then heated to 850-900°C with the result that the patterns are completely removed. The remaining cavities are filled with metal at normal pressure or under a pressure of 2-3 atm.

This method is used to cast small- and medium-size parts of arbitrary shape. The high dimensional accuracy ($\pm 2^0/_{00}$) and surface finish (up to the 7th class) in many cases make it possible to dispense with subsequent machining, and for this reason this method is frequently applied for making parts from difficult-to-machine materials (for example, turbine blades from heat-resistant alloys).

Cavityless (full-form) casting. Patterns of foam polystyrene (density 0.01-0.03 kgf/dm^3) are moulded into unsplit sand moulds. When metal is poured in, the patterns are gasified, the vapours and gases escaping through the overflows and ventilation holes. Sublimation (heating to 300-450°C without access of air) and dissolution of the pattern in dichloroethane or benzene are another two methods employed to remove the moulded patterns.

The full-form casting makes it possible to obtain accurate castings of practically any shape.

Pressure die casting. Metal is poured into permanent steel moulds under a pressure of 30-50 atm. This method is highly productive and ensures accurate dimensions ($\pm 1^0/_{00}$) and good surface finish (up to the 8th class), and generally does not require any further machining. The method is used for the mass production of small- and medium-size parts predominantly from easily fusible alloys (aluminium, copper-zinc, etc.). The moulding dies for steel and iron castings must be manufactured from heat-resistant steel.

Now we consider the most widespread method — sand mould casting. Many of the design rules for sand castings are also applicable to castings obtained by other methods.

3.1. Wall Thickness and Strength of Castings

The walls of cast members feature unequal strength in their cross section because of the different conditions of crystallization. The strength is the highest in the surface layer where the metal, as a result of the increased cooling rate, gets a fine-grained structure and where residual compressive stresses favourable for the strength develop. In the surface layer of iron castings there prevail pearlite and cementite. The core which solidifies at a slower rate has a coarse-grained structure with the predominance of ferrite and graphite. Dendritic crystals and shrinkage cavities and porosity often develop in the core.

The thicker the wall, the greater the difference in strength between

the core and the skin. For this reason an increase in the wall thickness is not accompanied by a proportional increase in the strength of the entire casting. The dependence of strength on the sample diameter is illustrated in Fig. 58.

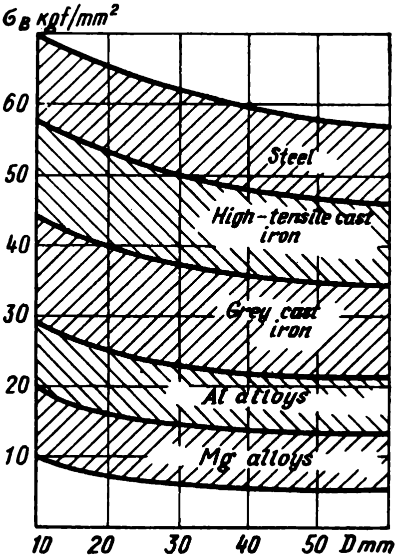


Fig. 58. Strength of casting alloys

For these reasons, and also to reduce weight, it is advisable to make the casting walls to the minimum thickness permitted by casting conditions. The required rigidity and strength should be ensured by ribbing, using rational profiles, and imparting convex, vaulted, spherical, conical and similar shapes to castings. This always results in lighter structures.

The quality of the shape of a casting may be approximately estimated by the ratio of its surface to the volume, or when the length is known, by the ratio of its perimeter S to cross section F

$$\Omega = \frac{S}{F} \tag{3.1}$$

Figure 59*a-c* specifies the values of Ω for several equivalent sections with different wall thickness. Massive shapes (Fig. 59*a*, *b*) are impractical as to their strength and weight. Thin-walled shapes greatly developed on the periphery (Fig. 59*c*) are the correct casting shapes.

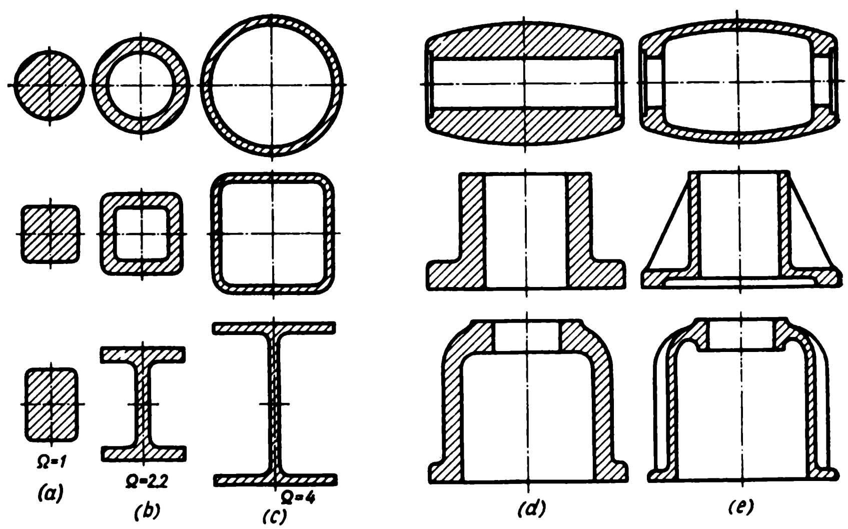


Fig. 59. Shapes of cast parts

Figure 59*d* shows irrational designs of cast parts in the shape of massive castings while their rational designs in the shape of thin-walled structures are presented in Fig. 59*e*.

The machining of cast parts should be minimized not only to reduce the manufacturing costs but also for strength considerations. The machining results in the removal of the strongest surface layer from the casting. The surfaces to be machined are reinforced by making the adjacent walls thicker.

3.2. Moulding

The design of a casting must ensure simple and convenient mould manufacture. This condition is broken down into the following particular ones:

(a) the pattern must be easily extractable from the mould;

(b) the cores must be easy to mould in core boxes;

(c) the shape and fastening of the cores must not hamper the assembly of the mould.

(*a*) *Elimination of Undercuts*

A pattern can easily be removed from the mould if its surface carries no *undercuts* — projections or recesses perpendicular or inclined to the direction of withdrawal — which are liable to cut off some portions of the mould when the pattern is extracted.

A scheme of undercutting is illustrated in Fig. 60*a*. The part has inclined ribs. When the pattern is withdrawn (the withdrawal

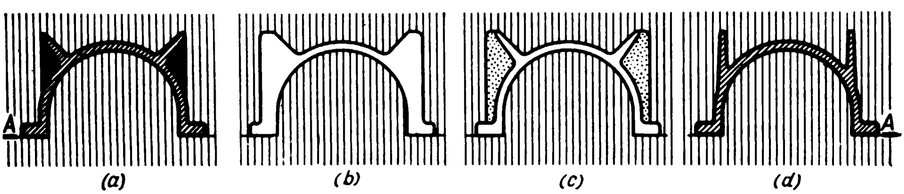


Fig. 60. Undercuts and their elimination

direction is shown by the hatching perpendicular to parting plane *A*-*A* of the mould) the ribs cut off the mould portions shown blackened in the drawing. The undercutting can be eliminated if the pattern portions hampering the extraction are made detachable or movable. Before the pattern is extracted these portions are taken away or drawn inside the pattern, after which the pattern can freely leave the mould. In another method the pattern is made so that the portions subject to undercutting are completely filled. This pattern takes the form shown in Fig. 60*b*. The required shape is obtained by

means of cores installed in the mould after removing the pattern (Fig. 60*c*).

All these methods make moulding more complicated and expensive. It is better to shape the part so as to exclude undercutting. When the ribs are parallel to the pattern withdrawal direction (Fig. 60*d*) the pattern can easily be taken out of the mould.

When designing a casting one must have a clear idea of the arrangement of the parting plane and the position of the part in the mould during pouring. As a rule, parts are cast with critical surfaces down,

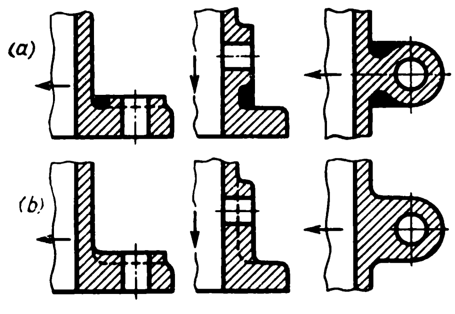


Fig. 61. Undercuts in moulding the bosses

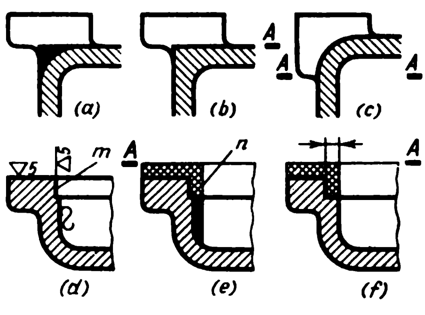


Fig. 62. Elimination of undercuts

since the metal in the lower portions of the casting is denser and better than in the upper portions. After establishing the parting plane, all the elements of the design must be inspected in succession and undercuts eliminated.

The *rule of shadows* is helpful in this case. Imagine that the part is illuminated by parallel rays normal to the parting plane (Fig. 60*a*). The shadowed portions show the presence of undercuts.

Figure 61*a* presents examples of undercuts when moulding bosses (the direction in which the pattern is extracted is shown by arrows). Figure 61*b* shows how the undercuts can be eliminated.

Examples of typical undercuts and the methods for their elimination are presented in Table 2.

Undercuts are not always seen clearly on drawings and can easily be overlooked by the designer. An example of an unapparent undercut is presented in Fig. 62*a* (the unit is shown in the moulding position; the parting plane is designated by the letter A).

The box fillet forms a dead volume (shown blackened in the drawing) in the bottom half mould.

This corner can be moulded if the vertical wall of the box is continued to the parting plane (Fig. 62*b*), or if the parting plane is transferred to the section where the fillet merges with the wall. In this case the lug must be extended to the parting plane (Fig. 62*c*).

Table 2

Elimination of Undercuts

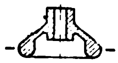
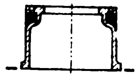
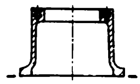
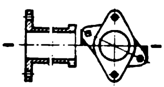
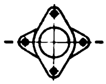
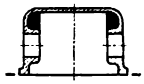
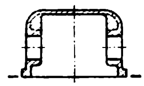
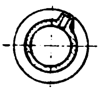
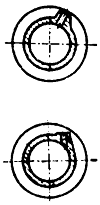
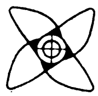
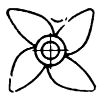
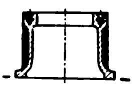
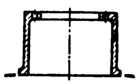
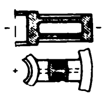
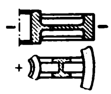
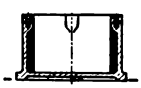
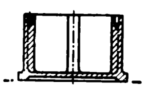
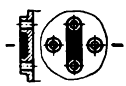
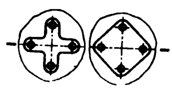
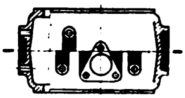
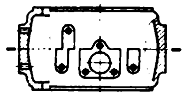
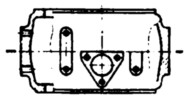
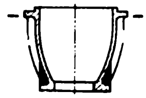
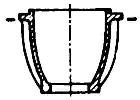
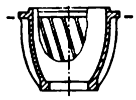
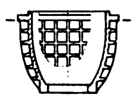
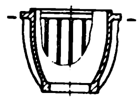
Original design	Corrected design and method of elimination	Original design	Corrected design and method of elimination
Handwheel		*Housing*	
	 Changing the shape of the part		 Eliminating the flange by changing over from the bolted to studded fastening
Pipe connection		*Housing*	
	 Arranging the flange axes at right angles		 Extending the bosses to the housing top
Tubular part		*Fan impeller*	
	 Changing the boss shape		 Eliminating the blade overlap
Housing		*Handwheel spokes*	
	 Enlarging the internal cavity of the housing		 Turning the spoke I-section through 90°
Housing		*Bosses*	
	 Extending the bosses to the housing bottom		 Merging the bosses together

Table 2 (continued)

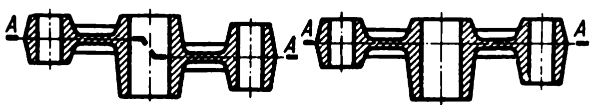
Original design	Corrected design and method of elimination	Original design	Corrected design and method of elimination
Housing with external bosses and pads		*Housing*	
	 Extending the bosses to the parting plane  Changing the arrangement of the bosses		 Eliminating the lower flange
		Housing with skew and crisscross ribs	
		 	 Changing over to straight ribs

In the cup-shaped part (Fig. 62*d*) the surface of recess *m* is too close to the adjacent rough wall.

The machining allowance *n* provided in the pattern (Fig. 62*e*) forms an undercut (blackened portion). The undercut can be eliminated, if the recess is deepened with respect to the rough surface by the machining allowance value (Fig. 62*f*).

(*b*) *Mould Parting*

The parting of moulds along inclined or stepped planes should be avoided as this complicates the mould manufacture.



(a) (b)

Fig. 63. Elimination of stepped parting of a mould

A stepped parting is required to mould a lever with offset arms (Fig. 63*a*). The moulding will be easier, if the arms are arranged in one plane (Fig. 63*b*).

The moulding of a curved pipe connection (Fig. 64*a*) can be simplified, if the connection axis is made straight, the position of the attachment points being slightly changed (Fig. 64*b*), or even kept unaltered, if necessary (Fig. 63*c*).

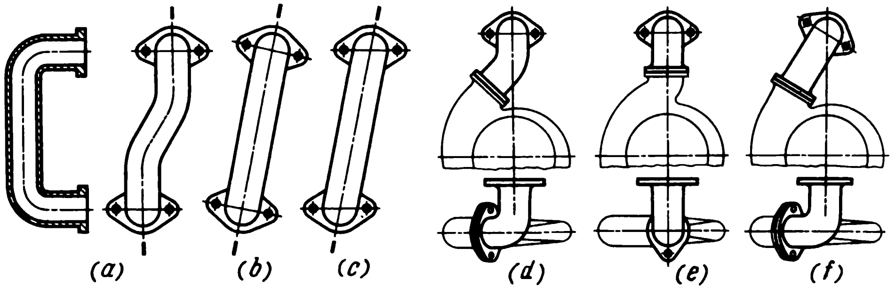


Fig. 64. Moulding of curved pipe connections

Figure 64*d-f* illustrates a change in the design of the outlet connection of a centrifugal pump. The most rational design is that in Fig. 64*f*, which simplifies the casting process and reduces the hydraulic losses in the pump because the fluid flow turns only once, and not twice as in the designs shown in Fig. 64*d* and *e*.

(*c*) *Open Castings. Cored Castings*

Open castings should preferably be moulded to patterns without the use of cores. In this case the pattern is shaped so as to conform

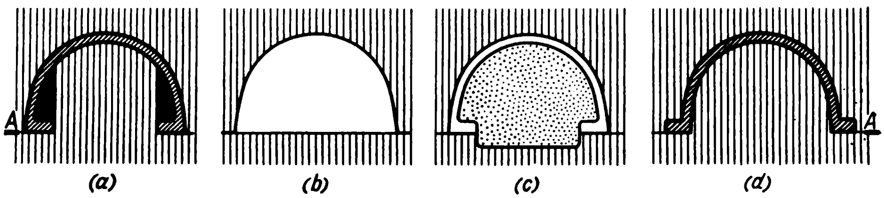


Fig. 65. Moulding of internal cavities

accurately to the shape of the final product. When a pattern is moulded a negative imprint of the cavity (*cod*) is obtained. This method can only be used if there are no undercuts on the internal surface of the part.

An example of internal undercutting is schematically shown in Fig. 65*a*. The part has a flange projecting into the cavity. When the pattern is removed the cod is damaged.

In the presence of internal undercuts the use of cores is the only way of moulding the cavity. In this case a solid pattern leaves in the mould the impression shown in Fig. 65*b*. The internal cavity is formed by means of a core (Fig. 65*c*).

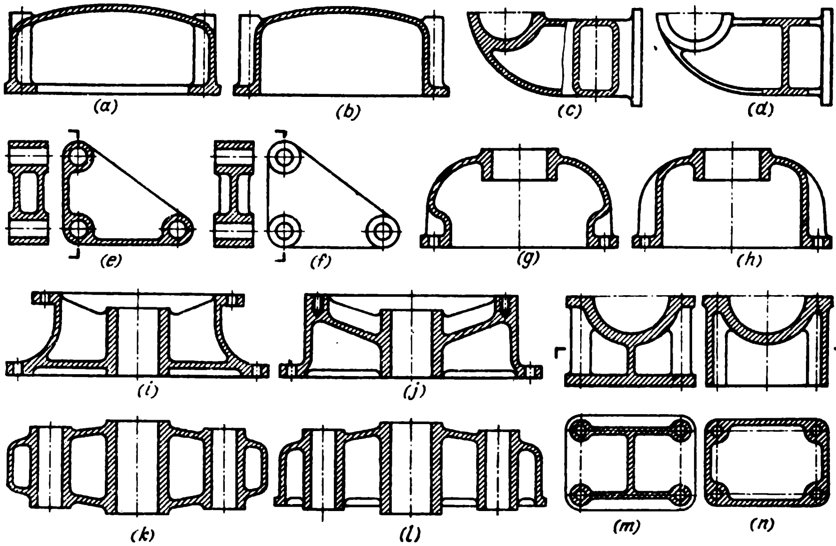


Fig. 66. Cored and coreless moulding
a, *b*—cover; *c*, *d*—bracket; *e*, *f*—lever; *g*, *h*—housing; *i*, *j*—adapter; *k*, *l*—rotor; *m*, *n*—bearing shell

The part can easily be modified to suit coreless moulding by placing the flange on the outside (Fig. 65*d*).

Examples of adapting standard parts to coreless moulding are illustrated in Fig. 66.

The requirements of simple and inexpensive production do not alway coincide with the demands for the proper strength and rigidity of parts and their convenient operation.

The open design of a cover (Fig. 66*b*) is simpler to manufacture than the design in Fig. 66*a*, which requires core moulding. But the design shown in Fig. 66*a* has a more attractive appearance.

The open design of a rotor (Fig. 66*l*) is simpler and can be made at a lower cost. But the box-like design shown in Fig. 66*k*, that requires the use of cores, is much stronger and stiffer.

In other cases, conversely, a less expensive design proves stronger and more convenient. Thus, the bearing body cast without cores (Fig. 66*n*) is stronger and more attractive than the one cast with cores (Fig. 66*m*).

The moulding of internal surfaces by means of cods is limited by the maximum permissible height of the latter. With the usual composition of moulding mixtures the height of bottom cods should be $H < 0.8S$ and that of the top ones, $h < 0.3s$ where S and s are the

mean cross sections of the cods, respectively (Fig. 67). In the case of reinforced moulds (moulds made from mixtures with bentonite or binders, skin-dried moulds, chemical-set moulds, etc.), and also when the moulding is done mechanically, the height of the cods can be increased by 30-50 per cent as against the above values.

The designs of cast elements should be devoid of narrow cavities, deep pockets of small cross section, etc. (Fig. 68*a*). Such cavities

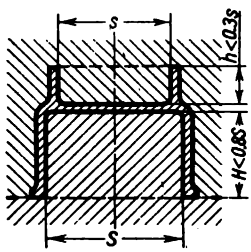


Fig. 67. Determining the height of cods

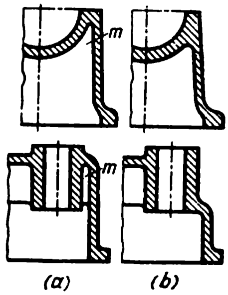


Fig. 68. Strengthening of weak elements of a mould

are poorly filled with the moulding mixture and form in the mould weak pillar- or band-type protrusions *m* which crumble when the pattern is extracted and are easily washed away by the pressure of liquid metal. The methods of eliminating these faults are illustrated in Fig. 68*b*.

(*d*) *Cores*

When, designing internal cavities the core should be given such a shape as ensures its easy extraction from the core box.

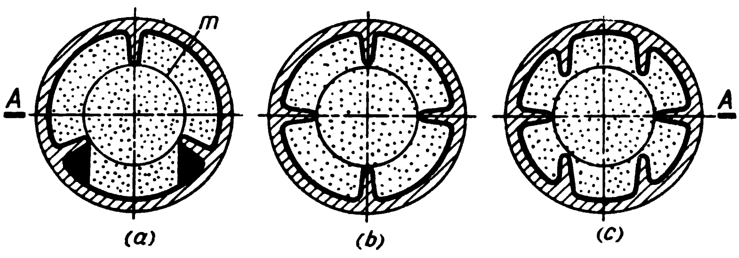


Fig. 69. Moulding a core

Figure 69*a* illustrates a core used to form in a part a cylindrical cavity with internal ribs. The shape of the core allows the box parting to be made in plane *A-A* only (because of the presence of annular rib *m* in the cavity). The ribs form undercuts in the box. In such

cases the cores have to be made up of separate parts bonded together, which complicates the core manufacture and reduces the casting accuracy. In the good designs shown in Fig. 69*b* and *c* the ribs are arranged in the parting plane or perpendicular to it, and the core is easily extracted from the box.

Particular difficulties arose when moulding cores for structures with skew axes. Figure 70*a* shows a manifold in the form of cylindrical header *m* with drop-shaped branches *n* the axes of which are offset with respect to the header axis.

In this design the core cannot be moulded. With any arrangement of the core box parting plane—horizontal (plane *A-A*, Fig. 70*b*), vertical (plane *B-B*,

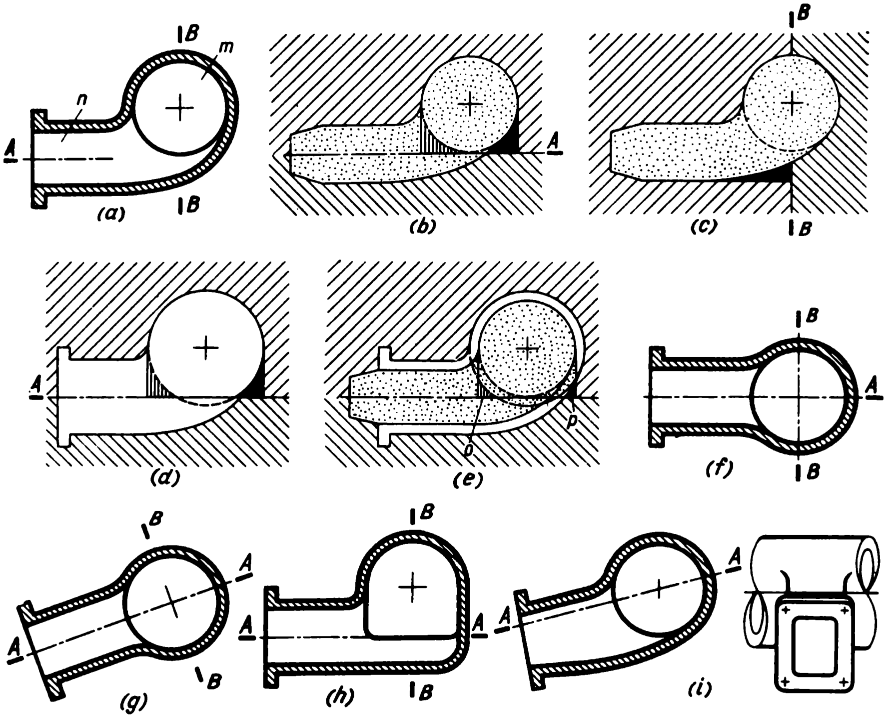


Fig. 70. Moulding a drop-shaped cylindrical header

Fig. 70*c*) or, the more so, inclined—undercuts are formed (shaded and blackened portions on the drawing).

Undercuts are also formed when the pattern is moulded into a mould parted along plane *A-A* (Fig. 70*d*). The mould cannot be assembled. The core makes it impossible to join the top and bottom half moulds (sections *o*, *p* in Fig. 70*e*).

Bringing the header and branch in line (Fig. 70*f*, *g*) makes it possible to mould the core in a box parted along plane *A-A* or *B-B*. The pattern can be moulded in if the mould parting passes through plane *A-A*.

If the skew axes are to be maintained the shape of the manifold should be changed according to Fig. 70*h*. In this case the core can be moulded if the core

box parting is in plane *A-A* or *B-B*, and the pattern moulded in if the mould parting is in plane *A-A*.

In the design shown in Fig. 70*i* the branches are given a rectangular cross-section. The core and the mould can be made with the parting along plane *A-A* or along any other plane passing along the branches and arranged within the limits of the straight portion of the side walls of the branch. In this case the manifold retains its assigned shape.

(*e*) *Installation of Cores in a Mould*

The shape of the internal cavities in a mould must permit the easy installation of cores. The design of the drop-shaped manifold in Fig. 70*e* is an example of a mould that cannot be assembled.

Figure 71*a* shows the head of an internal combustion engine with a spark plug well formed by a suspended core *1*. When assembling the mould, the core installed in the top half mould comes (in section *m*) against core *2* forming the water jacket of the head and installed previously in the bottom half mould.

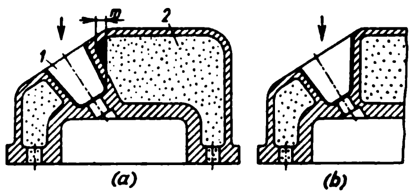


Fig. 71. Assembly of a mould

In the correct design (Fig. 71*b*) the well is shaped so that the top half mould can be easily mounted.

(*f*) *Escape of Gases*

The design of internal cavities should permit the escape of the gases evolved from the cores when the metal is poured in.

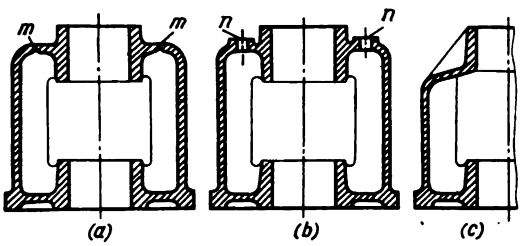


Fig. 72. Escape of gases from a core

An unsatisfactory design is illustrated in Fig. 72*a*. The gases accumulating in the upper part of the core form blowholes in sections *m*.

Provision should be made for holes *n* (stopped up afterwards) for the escape of gases (Fig. 72*b*). The vaulted shape of the upper portion

of the casting (Fig. 72*c*) ensures the escape of gases through the top core print.

Blowholes can be prevented by using core mixtures with a low gas formation.

(*g*) *Band Cores*

Slender cores are usually reinforced with a wire frame in order to increase their strength. When removing the core from the casting the frame has to be taken out, and this limits the minimum cross section of the core and calls for a well thought-out arrangement of holes for core prints.

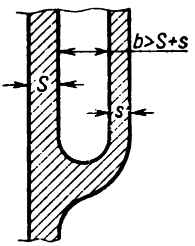


Fig. 73. Determining the minimum width of core-moulded cavities

For castings of small and medium size the thickness of cores reinforced with wire should be at least 6-8 mm. The core thickness can be reduced to 5 mm in local nicks.

The width of cavities should be not less than $b = S + s$ where S and s are the thicknesses of the walls forming the cavity (Fig. 73). It is better to make the cores as thick as the overall dimensions of the castings permit it.

(*h*) *Unification of Cores*

When designing castings with several cores of about the same shape, it is advisable to unify the cores in order to shorten their type list.

An example of the unification of cores for the crankcase of an in-line reciprocating engine is shown in Fig. 74. In the design in Fig. 74*a*

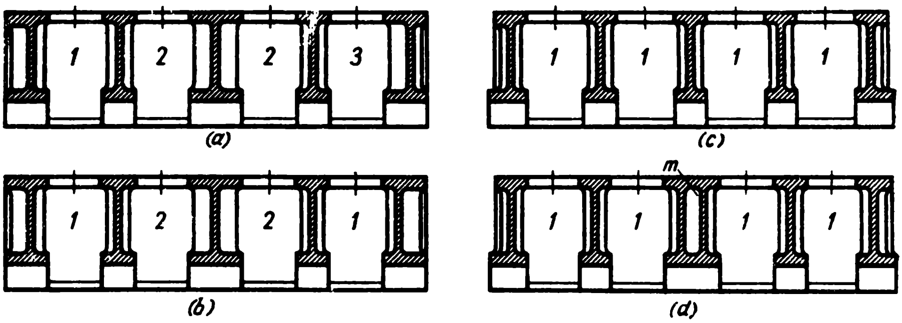


Fig. 74. Unification of cores

the internal cavities of the crankcase are formed by cores *1*, *2* and *3* of three different types. A slight change in the shape of the rear crank-case wall (Fig. 74*b*) makes it possible to reduce the number of the core types to two (*1*, *2*).

The number can even be reduced to one (Fig. 74*c*). However, this entails the shortening of the middle crankshaft bearing which in engines of this type is loaded more heavily than all the other bearings and therefore must be longer than they are.

In the design in Fig. 74*d* all the large cores are unified and the middle bearing is made longer by means of an additional core *m* that imparts a box-like shape to the middle crankcase partition.

(*i*) *Fastening of Cores in a Mould*

In castings with open lower cavities the cores are installed with their bases in the bottom box (Fig. 75*a*). The cores forming the upper cavities are suspended in the top box from an inverted cone (Fig. 75*b*)

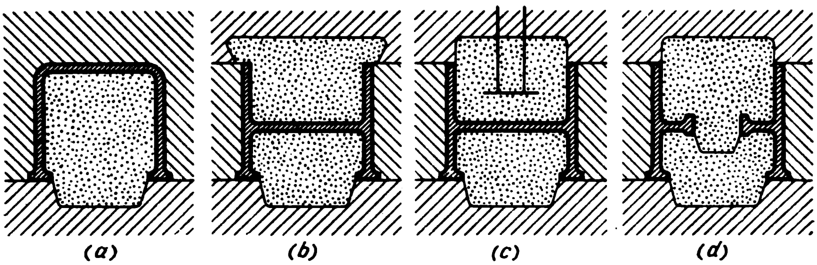


Fig. 75. Installation of cores

or from a wire (Fig. 75*c*) attached to a bar resting against the box. It is good practice to make the top core rest upon the bottom one through a hole in the horizontal wall of the casting (Fig. 75*d*).

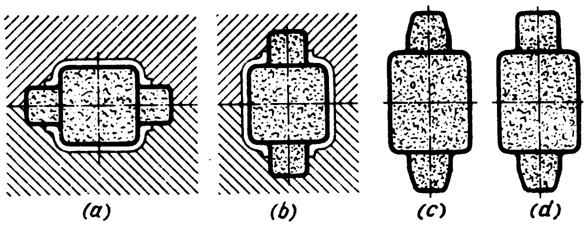


Fig. 76. Core prints

In closed cavities the cores are secured on *core prints* which take the form of projections moulded integral with the cores and installed in the seats made in the mould by the respective projections on the pattern. To make the mould assembly possible the core prints are arranged either in the mould parting plane (Fig. 76*a*) or perpendicular to it (Fig. 76*b*).

Core prints may be cylindrical (Fig. 76*a*, *b*) or conical (Fig. 76*c*). Conical core prints ensure a more accurate installation of the cores in the transverse direction, but their axial location is less definite than with cylindrical core prints which rest with their end faces against the core seats in the mould. Combinations of cylindrical and conical core prints (Fig. 76*d*) are frequently employed, the former being mounted with their flat end face supported in the direction of the axial force acting on the core during the metal pouring process.

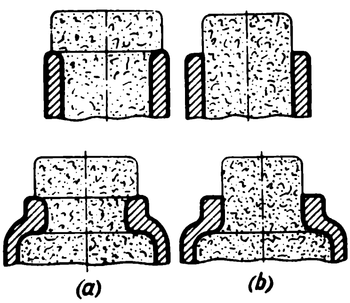


Fig. 77. Shapes of core prints

To simplify the core manufacture it is recommended to avoid fillets on the edges of holes in castings (Fig. 77*a*) and make the core prints plain (Fig. 77*b*).

Core prints are usually fixed in the holes available in the casting. In castings with closed internal cavities the cores are fastened by means of special prints brought out through holes in the casting walls. In the finished product these holes may remain open, if the functional purpose of the part permits it. The holes that impair the external appearance of the part, and also those in cavities which must be hermetically sealed, are stopped up.

To improve the fastening stability of cores and facilitate their knockout, the holes for the core prints should be made as large as it is permissible without materially weakening the casting and impairing its external appearance.

Core prints should be arranged so as to provide for the stable and, as far as possible, accurate location of the core in all the three coordinate planes. The fastening should be strong enough to endure the weight of the core and resist, during the pouring process, the dynamic action of the liquid metal stream and the hydrostatic forces that cause the core to rise due to the difference in the specific weights of the metal and the core material. In practice the hydrostatic force is most important.

The hydrostatic buoyant force acting on a core in a liquid metal is

$$P = V(\gamma_m - \gamma_c) \tag{3.2}$$

where V = volume of the core
γ_m and γ_c = specific weights of the metal and the core material, respectively

Let the volume of the core be 15 dm^3; γ_m, 7.4 kgf/dm^3 (molten iron) and γ_c, 1.4 kgf/dm^3 (skin-dried core). The hydrostatic force $P = 15\ (7.4 - 1.4) \approx$ ≈ 100 kgf, i.e., it exceeds the weight of the core ($G = 1.5 \times 1.4 = 21$ kgf) by about five times.

To prevent the core rising the core prints must be made to abut against the top half mould.

Cores must never be installed with a large overhang with respect to the fastening point (Fig. 78*a*) because the hydrostatic forces tend to twist the core out of its seat. Such cores should be secured at two points (Fig. 78*b*).

The core for a curved pipe (Fig. 79*a*) under the action of the hydrostatic forces applied to its buoyancy centre turns about its prints, as if about an axis. An additional support in the form of core print *1* should be provided on the bent portion of the pipe (Fig. 79*b*).

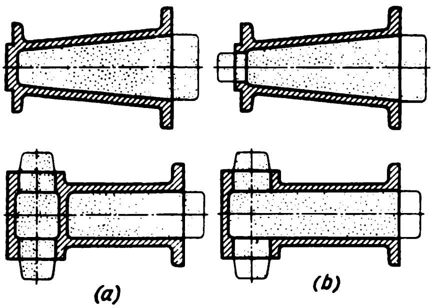


Fig. 78. Fastening of cores

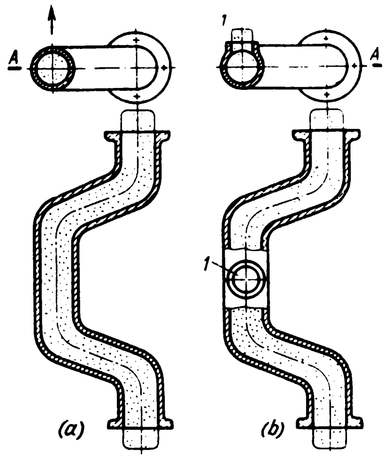


Fig. 79. Fastening of a core for a curved pipe

Sometimes, cores are secured against sagging, rising and lateral displacement with the aid of *chaplets* made in the form of metal cramps or with heads one of which is pressed against the mould and the other against the core. During pouring the chaplets fuse to the metal. For iron and steel castings these are made of steel, and in the case of non-ferrous castings, of the same metal as the casting.

The use of chaplets disturbs the homogeneity of the wall metal and reduces the castings strength. Chaplets must never be employed in cavities requiring hermetic sealing.

The fastening of a solid cylindrical core forming a cavity in a cylindrical housing is illustrated in Fig. 80. The core prints arranged in the parting plane of the mould (Fig. 80*a*) do not allow the gases to escape from the core. When the core prints are placed in the bot-

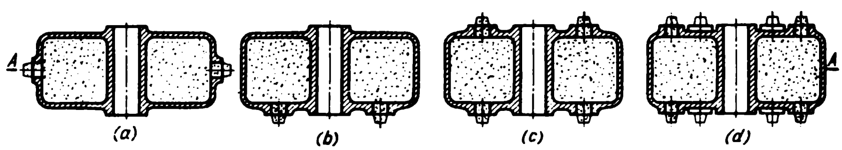


Fig. 80. Arrangement of core prints

tom half mould (Fig. 80*b*) the core is not secured against rising and also there is no escape for the gases. In the correct design shown in Fig. 80*c* the core is held by core prints in all directions, the upper prints ensuring at the same time proper core ventilation.

To secure the core reliably, provision should be made for several paired core prints along the periphery (best of all, three pairs of prints). The core mixture can easily be knocked out, if the prints

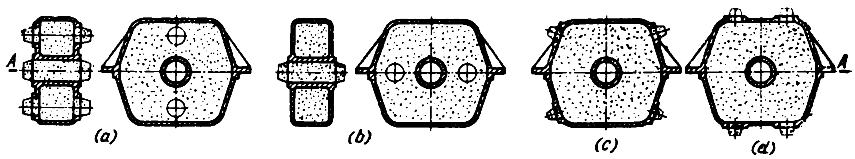


Fig. 81. Arrangement of core prints

are arranged in pairs along the same axis. For parts with very long cores the holes for the core prints should preferably be arranged in a staggered order (Fig. 80*d*).

Core prints should not hamper the mould assembly. Figure 81 shows a housing with an internal cavity formed by a core. When the core prints are positioned as shown in Fig. 81*a*, it is practically impossible to assemble the mould. In the correct design (Fig. 81*b*) the prints are arranged in the parting plane.

Figure 81*c*, *d* shows the arrangement of the core prints on the side walls.

The mould cannot be assembled when the prints are placed at an angle to the parting plane (Fig.81*c*).With the correct arrangement the prints are perpendicular to the parting plane (Fig. 81*d*).

(*j*) *Holes for Core Prints*

The edges of holes for core prints are as a rule reinforced with collars to compensate for the reduced wall strength. In iron castings the collars prevent the chilling of cast iron caused by the rapid cooling of the hole edges.

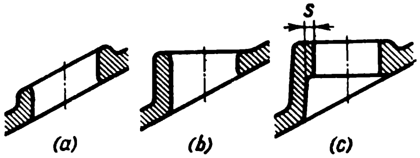


Fig. 82. Casting holes

The planes where the core print contacts the core and also the planes where the print passes into its seat in the mould should preferably be perpendicular to the print axis. Figure 82*a*, *b* shows a wrong and Fig. 82*c*, correct arrangement of the holes in inclined walls.

To ease the manufacture of core prints and prevent the weakening of the casting walls in the case of an accidental displacement of the

prints, the holes for the prints should be removed from the nearest walls to a distance $s \approx 4\text{-}5$ mm (Fig. 82*c*).

The methods of stopping up the core print holes are illustrated in Fig. 83.

Cylindrical holes of small diameter (up to 60 mm) are stopped up with threaded plugs (Fig. 83*a-f*).

Tightness is attained by installing gaskets (Fig. 83*a* and *d*), using taper threads (Fig. 83*b* and *c*) or an incomplete thread tightened

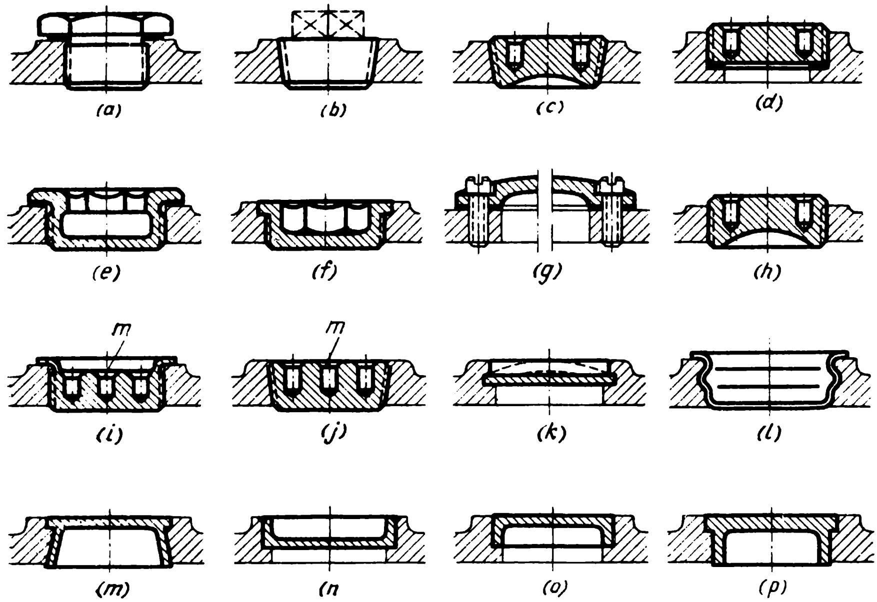


Fig. 83. Methods of stopping up the casting holes

until the last thread is forced into the threaded hole (Fig. 83*h*). The threads are coated with sealing compounds. Heat-resistant compounds (siloxane enamel) are used for parts operating under high temperatures.

To improve the external appearance, the tightening means on the plugs are usually made sunk (Fig. 83*c* and *d*). The tightening hexagons and tetrahedrons are cut flush with the hole edges after screwing the plugs home (Fig. 83*b*).

Large-size or shaped ports are closed with bolted plates or cast covers (Fig. 83*g*).

Screw plugs are locked by embossing or flaring (Fig. 83*i*).

In castings made of plastic metals (steel and non-ferrous casting) the plugs are secured by rolling in the casting surface (Fig. 83*j*). Centre holes *m* must be provided in the plugs to centre the rolling tool.

Spherical deformable plugs (Fig. 83*k*) are made of plastic low-carbon steel. During installation the plug is flattened and its edges cut into the walls of the hole, forming a strong and tight seal. Use is also made of plugs flared either from the outside (Fig. 83*l*) or from the inside (Fig. 83*m*). In steel members the plugs are fastened by soldering or welding (Fig. 83*n*). The plugs can also be fixed in place with epoxy adhesives (Fig. 83*o* and *p*).

Out of all these methods preference should be given to designs requiring minimum machining, for example, in the case of small holes, to screw plugs with a taper thread.

A smooth surface is very important for the holes arranged on the outside. Recesses and pockets which accumulate dirt are undesirable (Fig. 83*e*, *f*, *l* and *n*). In this respect the designs in Fig. 83*m*, *o* and *p* are preferable.

3.3. Simplification of Casting Shapes

The shape of castings should be simplifed to reduce the costs of production and increase the casting accuracy. The outlines of parts

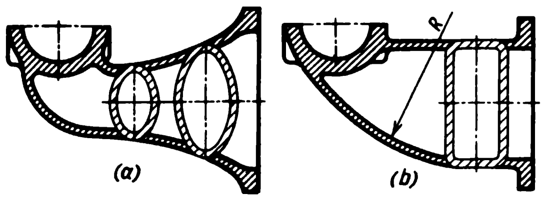


Fig. 84. Simplification of casting shapes

and inner cavities should be formed by simple straight lines, circular arcs, etc.

The bracket shown in Fig. 84*a* has unreasonably intricate profile and cross section. The transitions between cross sections are complex, and it is difficult to maintain identical the transitions in the pattern and core box. The walls of the casting will therefore inevitably differ. In the practicable design shown in Fig. 84*b* the cross sections have a simple rectangular shape.

3.4. Separation of Castings into Parts

It is good practice to separate large and intricate castings into parts.

Because of the convex bottom the housing of a vertical reduction gear (Fig. 85*a*) requires casting into a mould with cover cores. Upon separation (Fig. 85*b*) the parts of the housing take the shape of simple open castings moulded without cores.

Figure 85*c*, *d* shows an example of separating a framed bed into plain frames of simple shape.

In units consisting of several cast parts it is advisable to simplify the most intricate and largest casting while slightly complicating

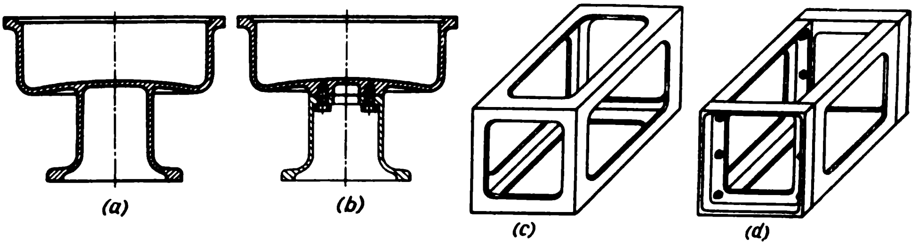


Fig. 85. Separation of castings

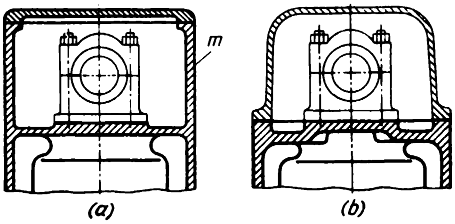


Fig. 86. Simplification of castings

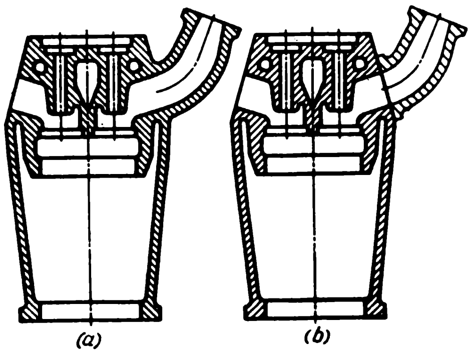


Fig. 87. Simplification of castings

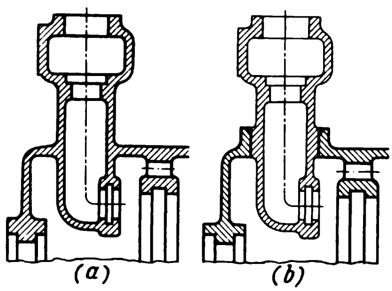


Fig. 88. Solid-cast (*a*) and welded-cast (*b*) designs

the simpler ones. In the design of the cylinder block of an internal-combustion engine (Fig. 86*a*), integrating the walls *m* with the cover (Fig. 86*b*) simplifies the casting and machining of the block and eases the access to the valve mechanism.

The projecting parts of housing-type components (Fig. 87*a*) should preferably be made detachable (Fig. 87*b*).

Figure 88*a*, *b* shows an example of simplifying a steel casting by using a welded-cast design.

3.5. Moulding Drafts

To facilitate the extraction of the casting pattern from the mould the pattern surfaces perpendicular to the mould parting plane are given the so-called *pattern drafts* (*tapers*).

Table 3

Standard Pattern Drafts

Height h above mould parting plane, mm	Pattern draft angle α	Draft ($\tan \alpha$)	h, $\tan \alpha$, mm
Up to 20	3°	0.0520	Up to 1
20-50	1°30′	0.0260	0.5-1.25
50-100	1°	0.0175	0.9-1.80
100-200	45′	0.0130	1.3-2.60
200-800	30′	0.0100	2.0-8.00
800-2000	20′	0.0060	5.0-12.0
Over 2000	15′	0.0040	Over 8

Table 3 presents standard pattern drafts, depending on the height h of the tapered pattern surface above the mould parting plane, and the corresponding transverse displacement $h \tan \alpha$ of the extreme points on this surface.

The values of standard pattern drafts are not indicated on drawings, and cast parts are drawn without such drafts. However, the drafts should be taken into account when designing castings of a large height in the direction perpendicular to the mould parting plane.

In a cylindrical part (Fig. 89*a*) the flange is machined to a diameter of 560 mm, i.e., it is 10 mm larger than the diameter of the rough surface (550 mm). This shape is impossible to produce because with a standard pattern draft of 1 : 100 the diameter of the rough surface at the base of the cylinder is $550 + 2 \times 750 \times 0.01 = 565$ mm and the tool cuts into the wall (Fig. 89*b*). It is necessary either to enlarge the diameter of the surface to be machined up to 575 mm, which entails increasing in the diameter of the bolt circle from 600 to 615 mm (Fig. 89*c*), or reduce the diameter of the upper portion of the cylinder down to 535 mm (Fig. 89*d*), if the flange shape is specified.

It is better to indicate the draft for large-size castings, or more preferable, to provide for *design tapers* that deliberately exceed the pattern drafts. It is not obligatory to adhere strictly to standard design tapers (Fig. 90). The shape of a part should be designed so as

to ensure its maximum strength and rigidity, and good external appearance as well, account being taken of the moulding, casting and machining conditions.

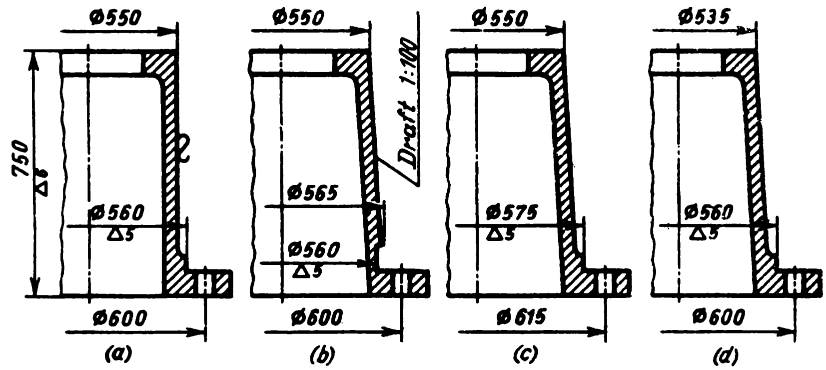


Fig. 89. Effect of casting drafts on the design

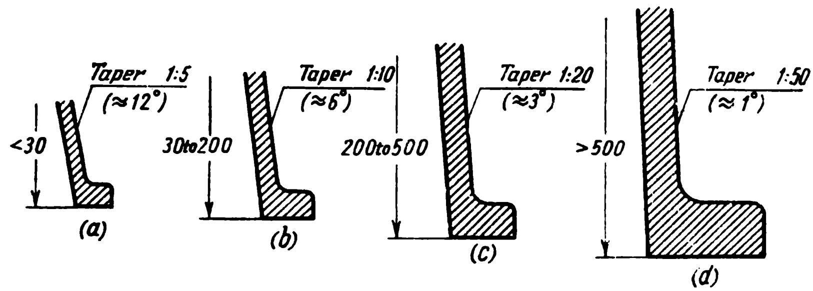


Fig. 90. Standard design tapers

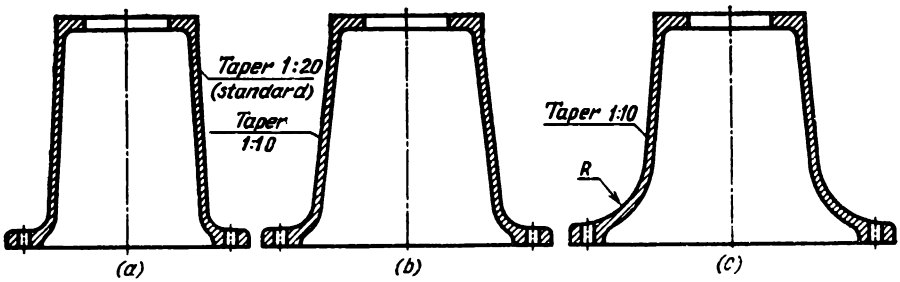


Fig. 91. Shapes of cast parts

Examples of shaping a cast part in the order of its increasing rigidity and improving its casting conditions are illustrated in Fig. 91*a*-*c*.

3.6. Shrinkage

Shrinkage is contraction in the size of a casting in cooling. *Linear shrinkage* (in per cent) is expressed as

$$\frac{L-L_0}{L_0}=\alpha\,(t_s-t_0)\,100\ \%$$

where L = size of the casting at temperature t_s of the metal solidification (solidus point)
L_0 = size after cooling to temperature t_0 in the premises
α = mean value of the linear (thermal) expansion coefficient of the metal within interval $t_s - t_0$

The value of the linear expansion coefficient, specific for each metal, somewhat diminishes as the metal temperature drops and changes in a step-like manner during phase transformations in the process of cooling (increase of volume in the pearlitizing of steel, pearlitizing and graphitizing of grey iron within the eutectoid transformation interval of 720-730°C).

Volume shrinkage characterizes the change (in per cent) in the volume of a casting in cooling. From the previous formula

$$\frac{V-V_0}{V_0}=\left(\frac{L}{L_0}\right)^3-1=[1+\alpha\,(t_s-t_0)]^3-1\approx 3\alpha\,(t_s-t_0)$$

i.e., volume shrinkage is about three times greater than linear shrinkage.

Shrinkage is one of the main casting properties of a material and, alongside other properties (castability, thermal capacity, heat conductivity, oxidability, liability to segregation), shows whether the given metal is suitable for casting.

The smaller the shrinkage, the higher the dimensional accuracy of the casting and the less the hazard of shrinkage stresses, cavities, cracks and warpage in the casting.

The linear shrinkage values for the main casting alloys are as follows:

Material	Linear shrinkage, %
Phosphoric iron	0.7-0.8
Grey iron	1.0-1.2
High-tensile cast iron	1.5-1.8
Carbon steel	1.8-2.0
Alloy steel	1.8-2.5
Phosphor bronze	0.6-0.8
Tin bronze	1.3-1.6
Aluminium bronze	2.0-2.2
Aluminium-copper alloys	1.[illegible]-[illegible].5
Aluminium-magnesium alloys	1.2-1.3
Aluminium-silicon alloys	1.0-1.2
Magnesium alloys	1.5-1.7

These figures refer to the case of *free shrinkage* determined from samples cast in open horizontal moulds. The actual shrinkage depends on the resistance to the contraction in the dimensions of the casting offered by the internal portions of the mould (*restricted shrinkage*). With rigid cores, shrinkage may be 30-50 per cent less than free shrinkage, but in this case higher shrinkage stresses develop in the casting walls.

The shrinkage of a casting is taken account of by correcting the dimensions of the mould, using for the manufacture of patterns and core boxes *shrinkage* (*patternmaker's*) *rules* with dimensions increased by the amount of shrinkage as compared with normal ones.

3.7. Internal Stresses

Internal stresses arise in the casting walls whose shrinkage is restricted because of the resistance of the mould elements or the action of the adjacent walls. Shrinkage cavities and porosity appear in those parts of the casting that solidify last, i.e., in thick and solid portions from which heat withdrawal is difficult (*hot spots*).

Increased internal stresses make the casting warp and may lead to the development of cracks.

In the course of time, internal stresses are redistributed and partly dispersed as a result of slow diffusion processes (*natural ageing*). After two or three years the part changes its original shape, which in precision machines (metal-cutting machine tools, for example) is impermissible.

Shrinkage stresses develop only during those stages of cooling when the metal loses its plasticity (within 500-600°C for cast iron and 600-700°C for steel). At higher temperatures the change in dimensions is compensated for by the plastic flow of the metal and the shrinkage manifests itself only in the thinning of the walls.

In the box-shaped casting of length L and width l (Fig. 92*a*) the internal partition (shown black in the drawing) cools at a slower rate than the horizontal walls. Assume that at the given moment the partition has a temperature t_1 corresponding to the temperature at which the metal passes from plastic into elastic state, and the walls have a lower temperature t_2 at which the metal is already elastic.

While cooling further, below t_1, the partition material hardens and, contracting, undergoes tension. Since the contraction occurs in two directions (along axes x and y), by the end of cooling, biaxial tensile stresses develop in the partition and compressive stresses of reaction, in the walls.

If, conversely, the partition temperature at the initial moment is below the temperature of the walls (Fig. 92*b*), by the end of cooling, biaxial compressive stresses will arise in the partition and tensile stresses, in the walls.

As a rule, the portions of a casting which cool first undergo *compression*, and those cooling later are subjected to *tension*.

Let us find the shrinkage stresses for the case when the partition cools later (see Fig. 92*a*), considering deformation along axis x only.

By the end of cooling, the partition would have shortened by the amount $\lambda_1 = \alpha l\,(t_1 - t_0)$ and the walls, by a smaller amount $\lambda_2 = \alpha l\,(t_2 - t_0)$, where l is the length of the walls along axis x, and t_0 is the final temperature. The dif-

ference

$$\Delta\lambda = \lambda_1 - \lambda_2 = \alpha l\,(t_1 - t_2)$$

determines the magnitude of the stresses in the casting.

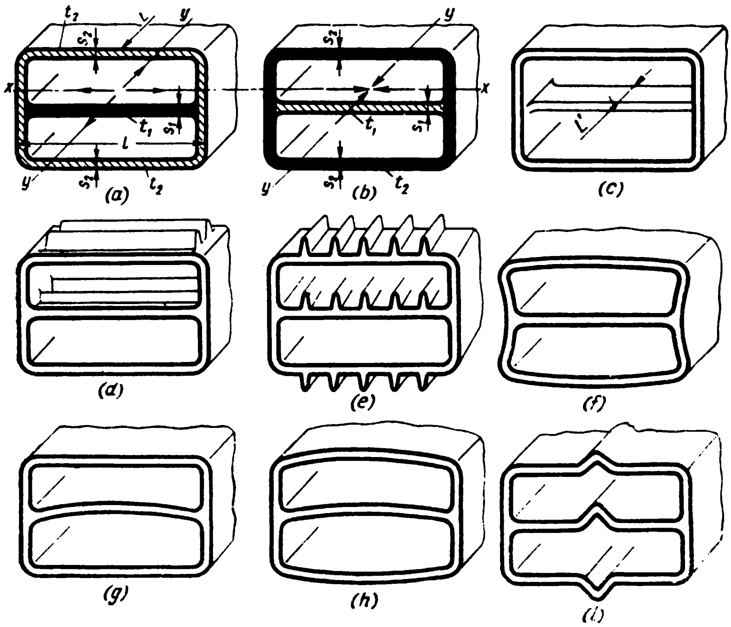


Fig. 92. Appearance of shrinkage stresses

According to Hooke's law

$$\Delta\lambda = \alpha l\,(t_1 - t_2) = \frac{Pl}{EF_1} + \frac{Pt}{EF_2}$$

where P = force developing in the system
E = mean modulus of elasticity within the temperature range $t_1 - t_0$
F_1 and F_2 = cross-sectional areas (normal to axis x) of the partition and the walls, respectively ($F_1 = s_1L$, $F_2 = 2s_2L$)

Force P is

$$P = \frac{E\alpha\,(t_1 - t_2)}{\frac{1}{F_1} + \frac{1}{F_2}}$$

The tensile stress in the partition

$$\sigma_1 = \frac{P}{F_1} = \frac{E\alpha\,(t_1 - t_2)}{1 + \frac{F_1}{F_2}}$$

The compressive stress in the walls

$$\sigma_2 = \frac{P}{F_2} = \frac{E\alpha\,(t_1 - t_2)}{1 + \frac{F_2}{F_1}}$$

The ratio between the stresses

$$\frac{\sigma_1}{\sigma_2}=\frac{F_2}{F_1}$$

These formulas show that the stresses are directly proportional to the product $E\alpha$ and the temperature difference $t_1 - t_2$ and depend on the ratio F_1/F_2 between the cross-sectional areas of the partition and the walls and do not depend on their length l.

To reduce the stresses in the partition, it is advisable to increase the thickness of the partition and reduce that of the horizontal walls. Danger can be expected from thin and narrow ($L' < L$) inner links (Fig. 92*c*) in which develop high tensile stresses (if they cool after the walls) or compression stresses (if they cool first).

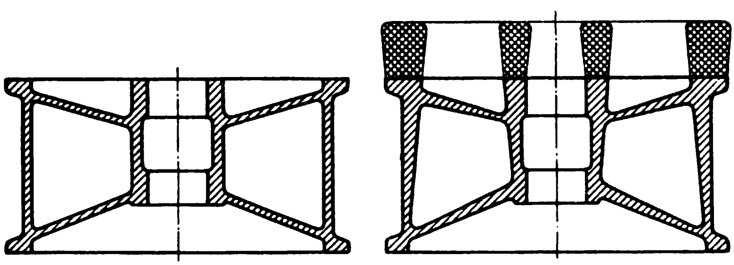
The magnitude and distribution of stresses can also be controlled by introducing ribs. It should be borne in mind that transverse ribs (Fig. 92*d*) only affect the shrinkage stresses acting along axis x, and longitudinal ones (Fig. 92*e*), along axis y.

The stresses cause the walls of castings to deform, as shown in Fig. 92*f* (the case of the partition solidifying after the walls). Their magnitude can appreciably be diminished, if the casting is made yielding in the shrinkage direction. For example, to reduce the shrinkage stresses acting along axis x, it is expedient to make the partition (Fig. 92*g*) or both the partition and the horizontal walls (Fig. 92*h*) curved, or to introduce shrinkage compensating buffers (Fig. 92*i*). The partition and the walls should be imparted a double-vaulted shape to decrease the shrinkage stresses acting simultaneously along axes x and y.

The primary cause of shrinkage stresses is the difference in temperature between the walls. With $t_1 = t_2$, the stresses are zero. It is on this principle that the method of *simultaneous solidification* is based. A casting can be freed of shrinkage stresses by making it cool uniformly, i.e., without any difference in temperature between the walls at each given moment.

3.8. Simultaneous Solidification

A combination of design and manufacturing measures is required to obtain simultaneous solidification.



(a) (b)

Fig. 93. Casting diagrams
a—simultaneous solidification; *b*—directional solidification

When designing castings on the principle of simultaneous solidification (Fig. 93*a*) the following rules should be adhered to:

(1) casting walls should preferably be of uniform thickness;

(2) casting elements cooling under conditions of reduced heat removal (internal walls) should have smaller cross sections to accelerate their solidification;

(3) transitions between casting walls of different thickness should be smooth;

(4) casting walls should have no abrupt changes but be connected by smooth transitions;

(5) local metal accumulations and massive elements should be avoided, if possible;

(6) sections where casting walls join massive elements should be gradually thickened towards the latter or reinforced with ribs.

It is good practice to increase the pliability of the casting in the direction of shrinkage deformations by introducing thermal buffers, making the walls vaulted, etc.

In practice, uniform cooling is ensured by active control of the cooling rate. Massive portions of a casting and also those from which heat removal is poor are cooled by means of metal chills and inserts made of heat-conductive moulding sands (moulding sand mixtures containing chromite, magnesite, etc.).

The formation of shrinkage cavities and porosity in massive portions is prevented by feeding molten metal to the parts which are the last to solidify (installation of ball gates, additional runners and risers, use of feeders).

The restriction of shrinkage by the inner mould elements is eliminated by employing pliable moulding sand mixtures, and porous, cellular and hollow cores.

Remaining stresses are removed by a stabilizing heat treatment. Iron castings are subjected to *artificial ageing* (soaking for 5-6 hours at 500-550°C with subsequent retarded cooling in the furnace). The castings are dressed-off prior to the ageing. Final machining is done after the ageing.

Parts subjected to artificial ageing hardly change their dimensions while in use.

An effective way of eliminating internal stresses and increasing the quality of castings generally is their *controlled cooling*. Metal is poured into heated moulds. After solidification (solidus point), the mould is slowly cooled, being held for some time at the phase transformation temperatures whereat the greatest volume changes take place, and also at those whereat the metal changes from plastic into elastic state.

This method eliminates the primary cause of shrinkage stresses because the temperature of all casting parts is the same at each given moment. The stresses due to the restriction from the mould are prevented by making use of pliable cores.

The heating of the mould before pouring removes from the moulding sand mixture moisture, vapours and gases which, when casting into cold moulds, cause vapour and gas cavities and porosity.

The cost of the process only slightly exceeds that of ordinary casting with subsequent stabilizing treatment.

3.9. Directional Solidification

This method is employed to cast parts from alloys with moderate casting properties. The wall sections are made to progressively increase from bottom upwards (see Fig. 93*b*). Solidification proceeds from bottom to top. While solidifying, the bottom sections are fed with molten metal from higher ones. The top sections which are the last to solidify are fed from massive risers on top of the casting. The transverse walls are inclined and grow thicker towards the top, and are connected with the adjacent walls by smooth fillets. The shrinkage cavity is formed in the riser. Non-metallic inclusions, slag, scabs and dirt go up into the riser.

The gradual movement of the solidification zone ensures correct shrinkage of the vertical walls. But the temperature differences in the vertical direction remain. The bottom horizontal elements of the casting that solidify first restrict the shrinkage of the top ones, and, as a result, tensile stresses develop in the top elements and compressive stresses, in the bottom ones. The shrinkage stresses reach their maximum in the top of the casting because of the considerable difference in cross section between the risers and the casting walls.

The shortcomings of the directional solidification method are as follows:

(1) increased casting weight due to the upward expansion of the walls (the shortcoming especially evident in high castings);

(2) great metal consumption;

(3) complicated moulding due to the presence of risers;

(4) difficult removal of the risers.

The method of directional solidification is predominantly used for steel castings, particularly when the weight of the part is of no great concern. The method is employed to cast (in a horizontal position) disk-type parts of small height (gears, pinions, diaphragms). For such parts the directional solidification principle consists in thickening the walls, making the disks conical, and increasing the transition fillets.

The simultaneous solidification method is preferable for intricate box-shaped parts.

3.10. Design Rules

(*a*) *Conjugation of Walls*

To provide for simultaneous solidification the thickness of the internal walls should be approximately equal to $0.8S$, where S is the thickness of the external walls.

The transitions from wall to wall should be smoothly curved (Fig. 94*b*). When walls are joined at right angles (Fig. 94*a*) the lines

of the heat flux meet in the inner corner of the joint and form a hot spot which slows down the cooling process. In addition, such joints make it difficult to fill the mould with metal and hamper shrinkage.

Figure 95*a-d* shows standard shapes of wall corner joints. With standard conjugation radii $R = (1.5 \text{ to } 2)\, s$ described from the same centre (Fig. 95*a*) the wall in the transition portion may be thinned if the core is displaced. Radii described from different centres make better connections. The outer radius is made equal from 1 (Fig. 95*b*) to 0.7 (Fig. 95*c*) of the inner radius. To improve heat removal, increase rigidity and prevent shrinkage cracks, conjugations of small radius should be provided with internal ribs (Fig. 95*d*).

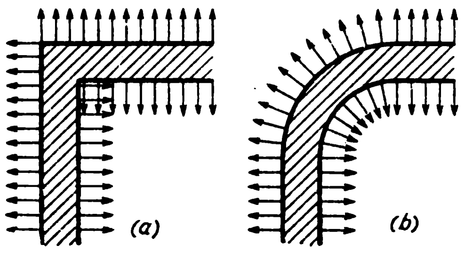


Fig. 94. Heat flux in a wall corner joint

Whenever the design allows, it is expedient to use the maximum transition radii permitted by the shape of the part (Fig. 95*e*).

Walls converging at an obtuse angle (Fig. 95*f*) are connected with radii $R = (50\text{-}100)\, s$. In such cases preference should be given to curved walls described by one large radius (Fig. 95*g*).

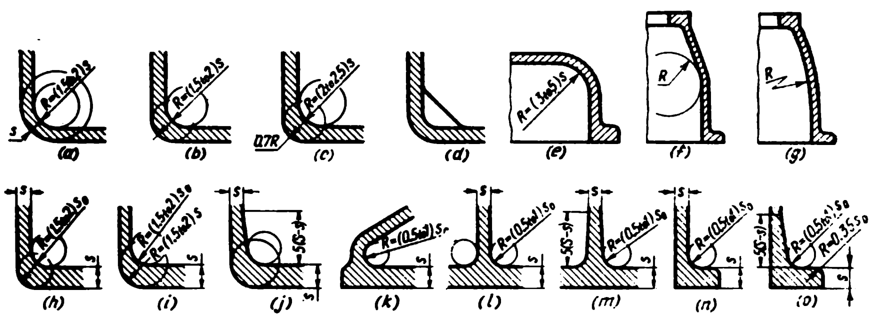


Fig. 95. Wall corner joints

The minimum conjugation radii of walls of different thickness may be found from the above ratios, having replaced s by the arithmetic mean $s_0 = 0.5\,(S + s)$ of the wall thicknesses (Fig. 95*h* and *i*). It may be adopted that $s_0 = S$ if the difference in the wall thickness is small.

Walls differing largely in cross sections should preferably be connected by a wedge-shaped portion of length $l \geqslant 5\,(S - s)$ (Fig. 95*j*).

Walls should never be connected at an acute angle (Fig. 95*k*). If this is inevitable, the conjugation radius should not be less than (0.5 to 1) s_0.

Figure 95*l*, *m* illustrates the recommended ratios for T-connections, and Fig. 95*n*, *o*, for connections of walls with flanges.

Walls of different thickness (Fig. 96*a*) should be connected by wedge-shaped transitions with tapers of from 1 : 5 to 1 : 10 (Fig. 96*b*

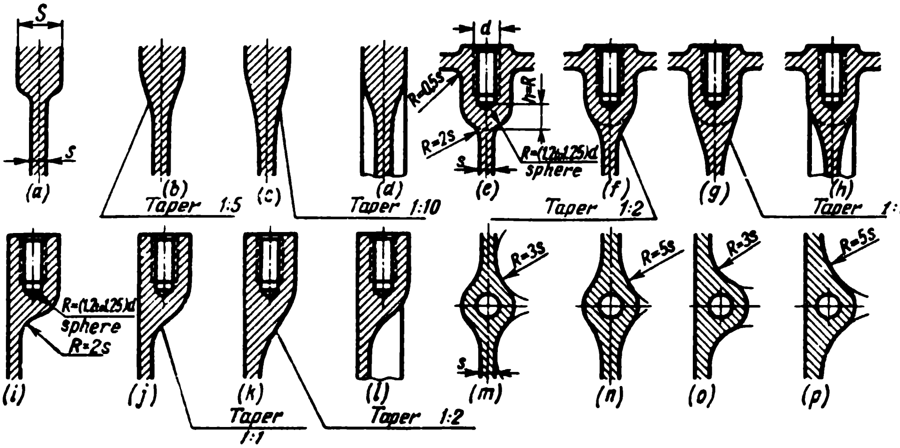


Fig. 96. Joints between casting sections of various thickness

and *c*). It is good practice to reinforce the transition section with ribs (Fig. 96*d*).

Figure 96*e*-*p* illustrates connection of walls with bosses. In the profile projection the bosses are linked with the walls by radii $R = 2s$ (Fig. 96*e* and *i*) or by tapers of from 1 : 1 to 1 : 5 (Fig. 96*f*, *g*, *j*, *k*) reinforced with ribs (Fig. 96*h*, *l*). In the plan projection, connection is made with radii $R = (3 \text{ to } 5)\, s$ (Fig. 96*m*-*p*).

The radii found from these tentative ratios are rounded off to the nearest standard dimension ($R = 1, 2, 3, 5, 8, 10, 15, 25, 30, 40$ mm). Since a slight change in the conjugation radii affects but little the quality of casting, it is recommended to unify these radii.

The predominant transition radius is as a rule not marked at each position on the drawing of a part, but is indicated in a drawing margin (or in specification) by an inscription such as: *Unspecified radii 6 mm.*

In the case of curved external corners the main radius is indicated by an inscription, such as: *Unspecified outer fillets R3.*

(*b*) Elimination of Massive Elements

Cast members should be free from local metal accumulations, and thick, massive elements forming hot spots. When designing a casting, one must carefully examine all places of material accumulation and

account for machining allowances which to a large degree affect metal distribution.

Figure 97 shows how massive elements (designated by the letter *m*) in a cast fastening flange (Fig. 97*a-c*), mounting pad (Fig. 97*d-f*), frame (Fig. 97*g-i*) and engine jacket (Fig. 97*j* and *k*) can be eliminated.

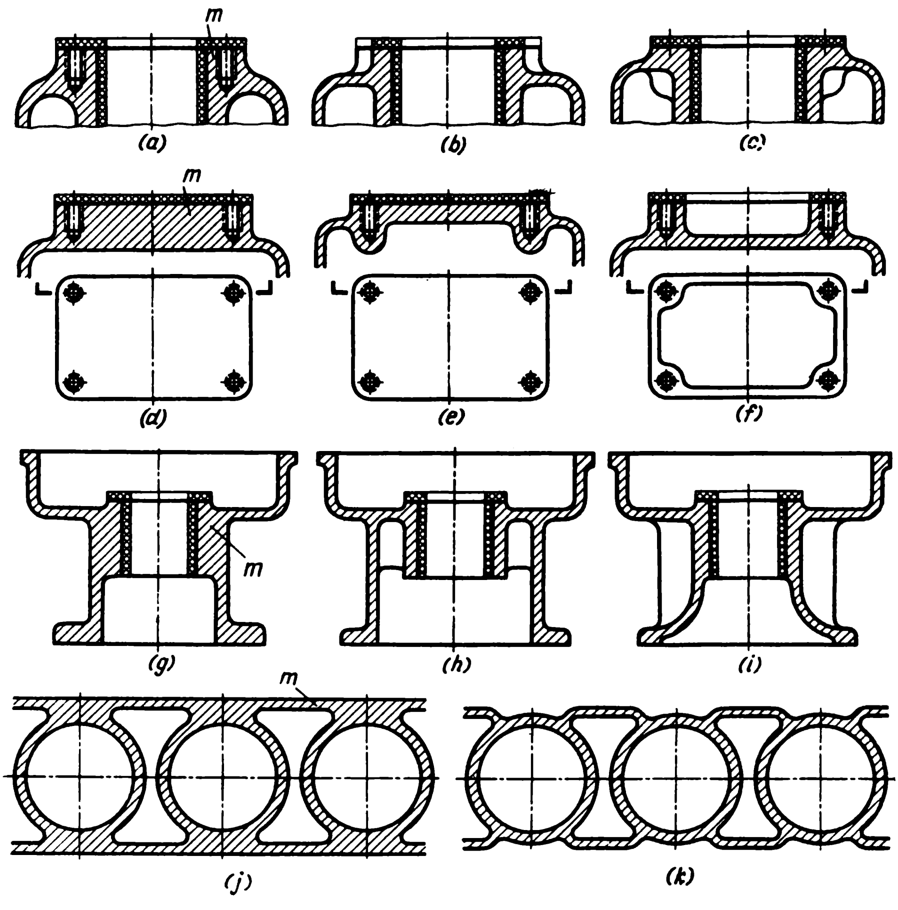


Fig. 97. Elimination of massive elements

Rapid cooling should be provided in the sections where massive elements are inevitable.

It is useful to enlarge the surface of contact with the moulding mixture by ribbing the walls. To improve the filling of the mould, the connection of massive elements with the nearest walls should be reinforced with fillets (Fig. 98*a*), wedge-shaped transitions (Fig. 98*b*),

bell mouthings (Fig. 98*c*) and ribs (Fig. 98*d*). It is advisable to use corrugated (Fig. 98*e*) and box-shaped (Fig. 98*f*) walls.

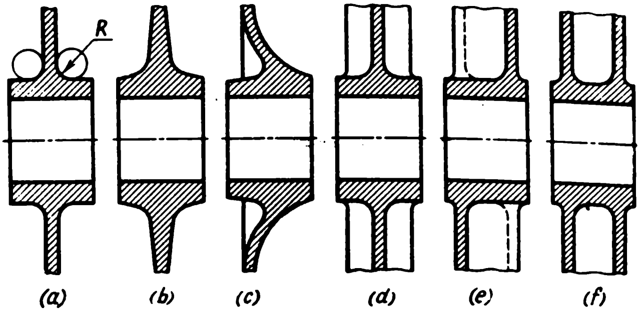


Fig. 98. Reinforcing the sections conjunct with bosses

These types of connection improve casting conditions and increase the rigidity and strength of castings.

(*c*) *Reduction of Shrinkage Stresses*

The shape of castings should facilitate shrinkage.

Figure 99 illustrates a large-diameter gear wheel whose rim is connected with the hub by spokes. The design with straight spokes

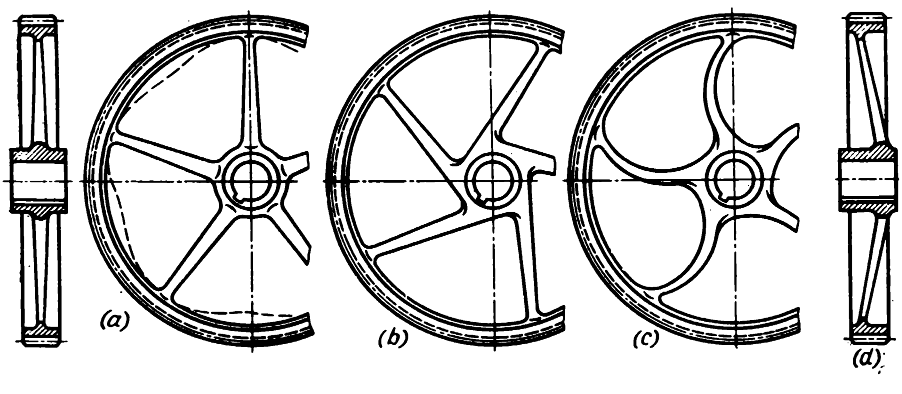


Fig. 99. Increasing the ductility of wheel spokes

(Fig. 99*a*) is wrong: the spokes solidify earlier and retard the shrinkage of the rim which is therefore subjected to a wave-like deformation. The internal stresses in such designs often cause the breakage of the rim.

It is more expedient to use tangential (Fig. 99*b*), spiral (Fig. 99*c*) or conical (Fig. 99*d*) spokes.

In a disk-type sheave with a massive rim (Fig. 100*a*) the disk solidifies before the rim and retards the shrinkage of the latter. Compressive stresses develop in the disk and tensile stresses in the rim.

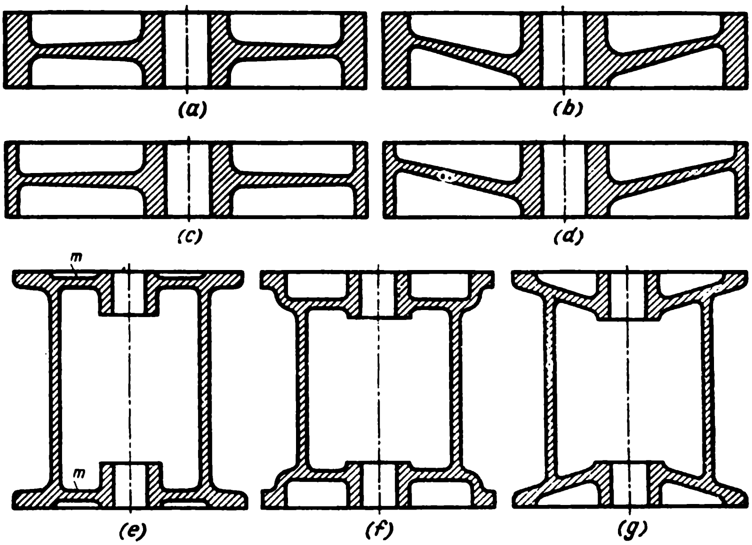


Fig. 100. Increasing the ductility of castings

If the rim solidifies first (Fig. 100*c*) the disk, while contracting, undergoes tension, and compressive stresses develop in the rim. In either case shrinkage stresses can effectively be diminished by making the disk conical (Fig. 100*b* and *d*).

In a cast frame (Fig. 100*e*) the partitions *m* located in one plane with massive flanges retard the shrinkage of the latter. The shrinkage conditions will somewhat be improved if the partitions are displaced from the plane where the flanges are arranged (Fig. 100*f*). But most advisable it is to make the partitions conical (Fig. 100*g*) or spherical.

Vaulted and convex shapes reduce shrinkage stresses, improve casting conditions, and increase the strength of parts because the resisting moments of cross sections of such shapes are greater. The rigidity of structures is also enhanced, which is especially important for castings made of alloys with a low modulus of elasticity (grey iron, light alloys).

(*d*) *Prevention of Blowholes*

The shape of a casting must provide for the rising of nonmetallic inclusions and the escape of gases which emerge as the casting cools down, because of their solubility in metal decreasing with temperature.

When a sump is cast with its bottom up (Fig. 101*a*) gas bubbles accumulate at the tops of the ribs and appreciably weaken them. It is better to make the bottom inclined and transfer the ribs onto

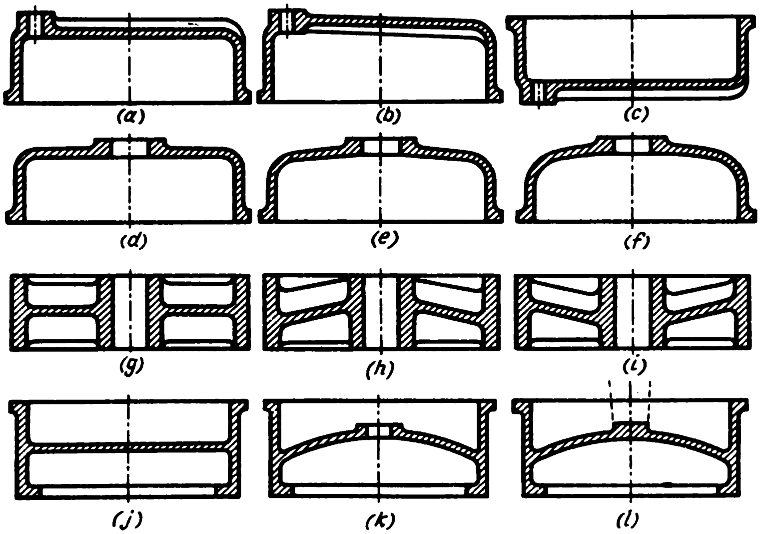


Fig. 101. Escape of gases

the internal surface (Fig. 101*b*). Such parts are recommended to be cast with the ribs down (Fig. 101*c*). In this case the blowhole porosity is concentrated in the riser on the flange, which is removed in subsequent machining. Casting with the inclination of the mould is likewise used.

For cylindrical parts (Fig. 101*d*) it is good practice to make the upper walls conical (Fig. 101*e*) or slightly spherical (Fig. 101*f*).

In disk-shaped parts (Fig. 101*g*) the disks and ribs should be inclined (Fig. 101*h*, *i*).

The internal partitions (Fig. 101*j*) should preferably be vaulted. Gas bubbles and nonmetallic inclusions can best of all be withdrawn by means of lugs (Fig. 101*k*) or bosses (Fig. 101*l*) in the upper part of the partitions, or with the aid of risers (dashed lines).

Casting under vacuum and addition of gas absorbing substances (cerium) to the casting metal are the process methods used to prevent blowhole porosity and cavities.

(e) *Rimming*

The external outlines of cast parts are usually rimmed (Fig. 102*a*, *b*) to obtain proper rigidity, uniform solidification and prevent chilling spots (in iron castings).

In parts joined by their end faces (Fig. 102*c*) the rims help to uniformly distribute tightening forces. With such rims it is much

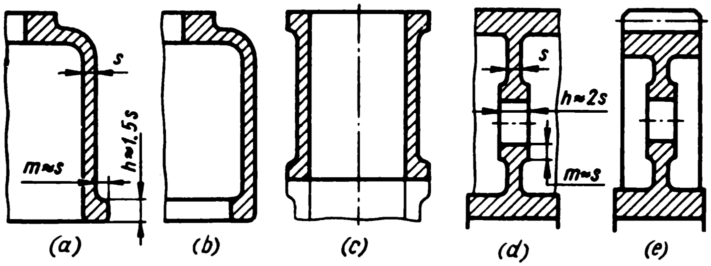


Fig. 102. Rimming

easier to remove irregularities and projections formed in joints due to inaccurate casting, and to match the outer contours of the joints.

As a rule, the lightening and process holes in the casting walls (Fig. 102*d* and *e*) should be rimmed to increase the strength of the casting and improve its cooling conditions.

Approximate dimensions of rims are given in Fig. 102*a* and *d*.

(f) *Flanges*

The thickness of flanges to be machined on one side (Fig. 103*a*) is made equal to (1.5 to 1.8) *s*, on the average, and that of flanges to be processed on two sides (Fig. 103*b*), (1.8 to 2) *s*, where *s* is the thickness of the adjacent wall.

In order to increase their strength and rigidity, flanges are connected with walls by ribs (Fig. 103*c*) or are

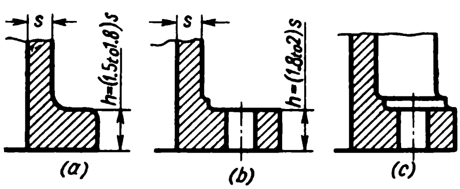


Fig. 103. Determining the thickness of flanges

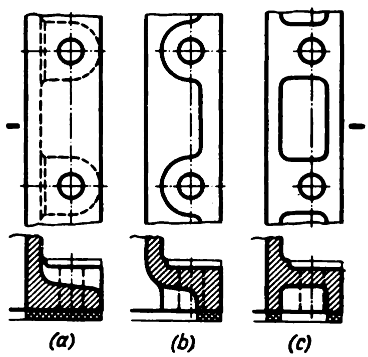


Fig. 104. Elimination of heavy sections in flanges

made box-shaped. Methods of eliminating heavy sections in thick flanges are illustrated in Fig. 104*a*-*c*.

(g) *Holes*

Long holes of small diameter should be avoided in castings.

The minimum diameter of holes in castings may be found approximately from the formula $d = d_0 + 0.1l$, where l is the length of the hole in mm (Fig. 105). For aluminium alloys and bronze $d_0 = 5$, for cast iron $d_0 = 7$ and for steel $d_0 = 10$ mm. Holes of smaller diameter are to be drilled. It is better to make long holes (such as oil ducts) by drilling or by casting-in tubes, or replace them with detachable pipelines.

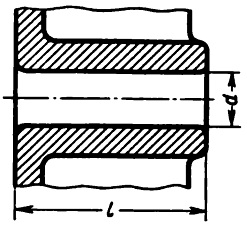


Fig. 105. Determining the diameter of holes in castings

The shape of cast oil ducts and cavities should allow their surfaces to be cleaned completely of burnt-on sand and other contamination. After careful cleaning the surfaces should be coated with oil- and heat-resistant compounds (bakelite or siloxane enamels).

(h) *Ribs*

Ribs are used to increase the rigidity and strength of cast parts and to improve casting conditions. A rational arrangement of ribs improves the feed to casting elements and prevents shrinkage cavities and internal stresses.

The shapes of ribs are illustrated in Fig. 106. Ribs arranged in a plane perpendicular to the mould parting should have casting drafts.

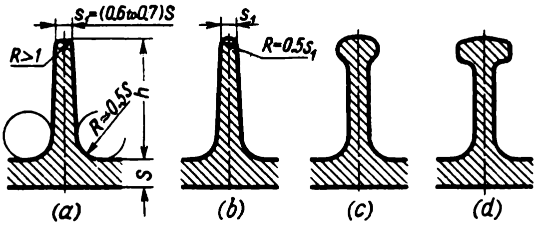


Fig. 106. Shapes of ribs

The thickness s_1 of the rib at the top is its basic dimensional parameter (Fig. 106a). For ribs 20-80 mm high the standard drafts in current use (see Table 3) give practically the same rib thickening of 2-3 mm towards the base (on both sides of the rib), the thickening being almost independent of the rib height.

Fillets with a radius of at least 1 mm are obligatory at the top of the ribs. The tops of ribs with a thickness of less than 6-8 mm are

rounded off with a radius $R = 0.5s_1$ (Fig. 106*b*). The base of the ribs is connected with the wall by fillets of radius $R \approx 0.5S$.

Bulb-shaped (Fig. 106*c*) and T-shaped (Fig. 106*d*) ribs are superior in strength. Such ribs are moulded with the aid of cores.

If a rib (Fig. 107*a*) solidifies in casting later than the wall (as is frequently the case with internal ribs), then during shrinkage (the shrinkage direction is shown on the drawing by dashed arrows) in the rib there develop tensile stresses (solid arrows). Conversely, if the rib solidifies first (Fig. 107*b*), it develops compressive stresses, which enhances the rib strength.

Faster cooling is effected by decreasing the rib thickness. The thickness of external ribs is usually taken at (0.6 to 0.7)S and that of

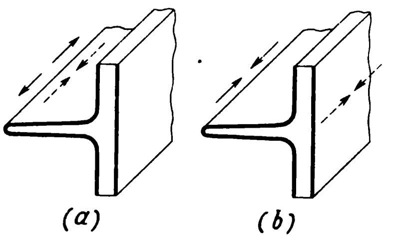


Fig. 107. Appearance of shrinkage stresses in ribs

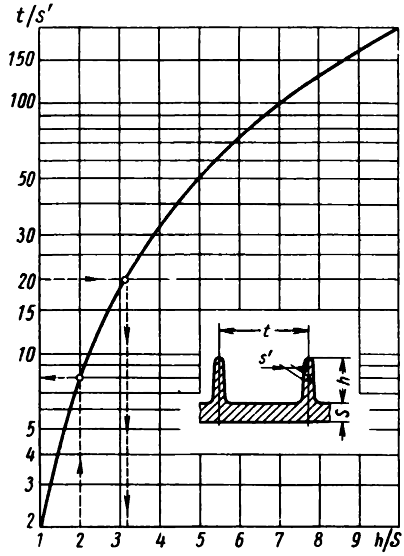


Fig. 108. Diagram for determining the maximum relative pitch of ribs t/s'

internal ones, allowing for the worse heat removal at (0.5 to 0.6) S, where S is the wall thickness (the upper limits refer to walls less than 10 mm in thickness and the lower ones, to those thicker than 10 mm).

Short, thin and sparsely distributed ribs with a small ratio of their total cross section to cross section of the wall reduce the resisting moment in bending of the section and strength of the part, although they increase its rigidity. This weakening can be avoided by distributing the ribs more densely. The maximum pitch at which no weakening occurs may be found from the formula

$$t = 2s'\left(\frac{h}{S}\right)^2 \tag{3.3}$$

where s' and h = mean thickness and height of the rib, respectively
S = wall thickness

The diagram in Fig. 108, drawn on the basis of Eq. (3.3), makes it easier to select the rib parameters.

1. Let the rib thickness be $s' = 5$ mm; $h/S = 2$. According to the diagram, the maximum allowable ratio $t/s' = 8$ and the maximum pitch $t = 8 \times 5 = 40$ mm.

2. Let the rib pitch be $t = 100$ mm; $S = 10$ mm; $s' = 5$ mm ($t/s' = 20$). From the diagram the minimum allowable ratio $h/S = 3.1$ and the minimum rib height $h = 3.1 \times 10 = 31$ mm.

Practically, ribs are made with a height of (2 to 6) S. Shorter ribs weaken the part without essentially increasing its rigidity, while taller ribs are difficult to cast.

Figure 109 illustrates examples of irrational and rational rib designs.

The design of the bracket in Fig. 109, *1* is unpracticable as the rib is subjected to tension. In design *2* the rib undergoes compression.

In the profile projection, ribs should have the simplest shapes. Concave ribs (design *3*) have the disadvantage of poor strength. When such ribs are subjected to bending and tension they develop high stresses proportional to the degree of concavity. Convex ribs (design *4*) are ugly in appearance and make the part heavier. Straight ribs (design *5*) are the best. They exhibit a high strength when subjected to tension or compression and bending.

In parts subjected to bending it is bad practice to connect the rib with the wall in the plane where the bending moment has a large magnitude (design *6*) because the resisting moment of the section in the plane AA where the rib merges with the wall is lowered. It is better to continue the ribs up to the part edge (into the region where the bending moment is less) and attach them to the belts of rigidity (design *7*).

Machining is liable to weaken the ribs and should be avoided. Design *8* of a plate with internal wafer-type ribbing is wrong. The ribs are brought out onto the plate surface to be machined, and the tops of the ribs will be shorn off during the machining. In the correct design *9* the ribs are arranged below the surface to be machined.

Measures should be taken to prevent undercutting of the ribs that adjoin surfaces to be machined. In designs *10* and *13* the ribs are arranged too close to such a surface. Deviations in the production conditions may be the cause of undercutting (designs *11* and *14*). The ribs should be positioned below this surface (designs *12* and *15*) by the amount $k = 3$-6 mm.

Ribs should never be extended to the rough surface of the flanges (design *16*) since moulding becomes difficult in the sections m where the ribs merge. It is advisable to arrange the ribs below the rough surfaces by an amount R equal to the radius of curvature of the flanges (design *17*).

The sections where the ribs pass into the body of the part (design *18*) should be described by radii R not less than 3-6 mm (designs *19* and *20*).

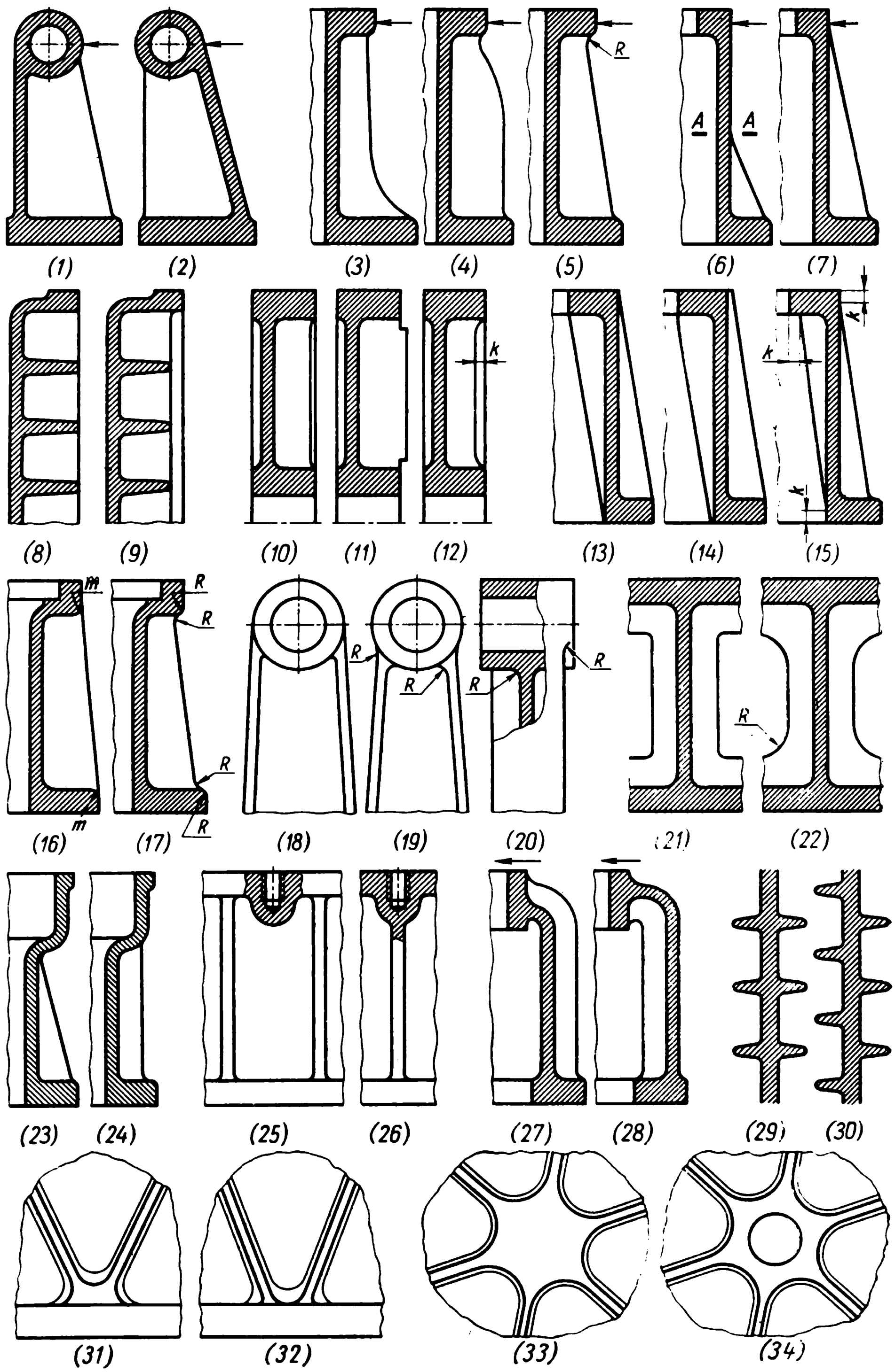


Fig. 109. Rib designs

Ribs connected (in plan projection) at an angle (design *21*) should have smooth transitions (design *22*).

As a rule, ribs should be brought to the rigidity nodes, i.e., sections where the wall directions change (design *24*) and fastening points (design *26*). Designs *23* and *25* are not recommended.

In shell-type parts (design *27*) subjected to bending internal ribs (design *28*) are preferable because in this case most of the bending

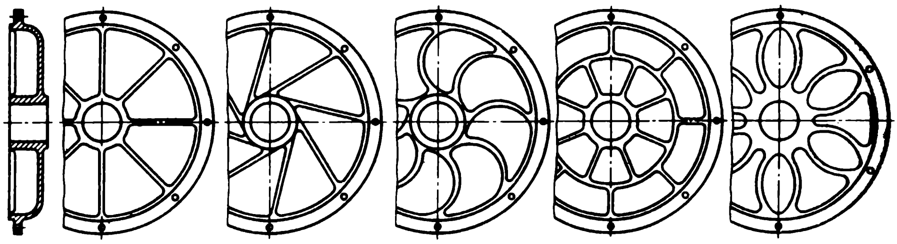


Fig. 110. Increasing the ductility of ribs

load is taken up by compressed ribs (on the side nearest to the direction of action of the bending force). Internal ribbing makes it possible to increase the radial size of the walls within the same overall

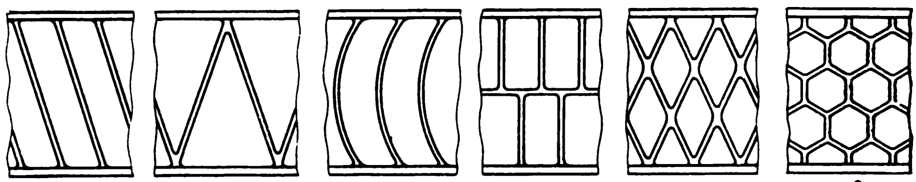


Fig. 111. Shapes of ductile ribs

dimensions and thus obtain a significant gain in rigidity and strength. This also improves the external appearance of the part and facilitates care of the product.

In the case of double-sided ribbing (design *29*) ribs should preferably be arranged in a staggered order (design *30*) to avoid local accumulations of metal and reduce shrinkage stresses.

Metal accumulations in sections where ribs join walls at an angle (design *31*) should be eliminated by spreading the ribs farther apart (design *32*). Heavy sections where several ribs meet (design *33*) are eliminated by providing lightening holes (design *34*).

In parts subjected to nonuniform heating when in operation, ribs undergo thermal stresses. If the walls of a part (Fig. 110*a*) are heated

more than the ribs, tensile stresses arise in the latter. Ribs with a temperature higher than that of the walls are compressed.

Thermal stresses can effectively be reduced if straight radial ribs (Fig. 110*a*) are replaced by tangential (Fig. 110*b*), spiral (Fig. 110*c*), sectional (Fig. 110*d*) and elliptic (Fig. 110*e*) ones.

Figure 111*a-f* presents types of ribbing of increased pliability. Such ribs can effectively be moulded only on flat or slightly curved surfaces parallel to the parting plane of the mould. It is difficult to mould such ribs on curvilinear surfaces and on solids of revolution.

(*i*) *Wall Thickness*

It is generally better to use walls of the minimum thickness permitted by the casting conditions and the strength of the part.

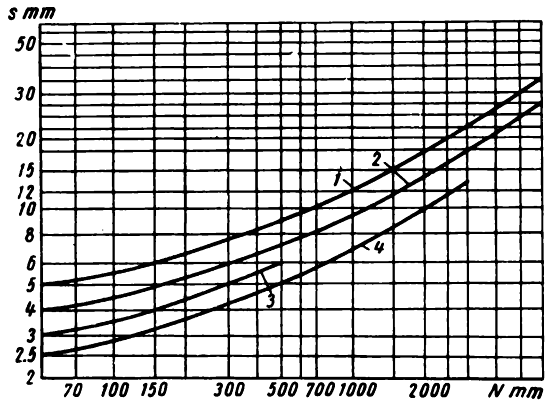


Fig. 112. Minimum wall thickness
1—steel; *2*—grey iron; *3*—bronze; *4*—aluminium alloy

Figure 112 illustrates the minimum wall thickness s for various casting alloys, depending on the reduced overall size of the part calculated from the formula

$$N = \frac{2l + b + h}{3}$$

where l = part length, mm
b = part width, mm
h = part height, mm

The diagram is plotted for external walls in sand-mould castings to the 2nd and 3rd grades of accuracy. The thickness of the internal walls, partitions and ribs is, on the average, 20 per cent smaller.

This diagram can only be used for the rough wall thickness estimates. The allowable wall thickness greatly depends on the casting shape. Intricate castings

moulded in several flasks with the use of a large number of cores should have thicker walls.

The casting process is as important: the composition of the moulding and core mixtures, feeding and cooling conditions, the design of the gating system, etc.

The wall thickness in heavily loaded parts (beds of hammers, stands of rolling mills, etc.) is determined by the magnitude of the acting loads and the rigidity requirements of the design, and considerably exceeds the values specified in Fig. 112. However, in this case too it is advisable to use walls of minimum thickness and ensure the required strength and rigidity by imparting rational shapes to the casting.

3.11. Casting and Machining Locations

The casting (*rough*) *location* is a surface or an axis with reference to which the initial machining operation is performed.

The rough surface location (*locating surface*) is a rough (as-cast), sufficiently long surface parallel or perpendicular to the *machining location*, i.e., the surface to be machined first. The shape of the rough locating surface should allow convenient and stable fastening of the part ready for machining, and tightening over this surface must not deform the cast blank.

A work surface must never be selected to serve as a rough location.

In the part shown in Fig. 113*a*, the rough location may be either the flange surface marked by a blackened lozenge, or the upper plane of the part (Fig. 113*b*). The machining location is indicated by a light lozenge.

The rough location is used to coordinate all the other rough surfaces of the casting (dimensions h), and the machining location, all the other work surfaces (dimensions h'). The machining location is designed with the minimum machining allowance so that allowances are uniformly distributed among the remaining work surfaces.

Sometimes, rough locations have to be specially provided by introducing process lugs (Fig. 113*c*) or by changing as required the shape of the part (Fig. 113*d*).

In the general case there must be three rough locations, one for each axis of the three-dimensional coordinate system used.

Axial locations (*locating axes*) are the axes of holes in the bosses. An axial location determines casting dimensions in the plane perpendicular to this axis, and a surface location, along the axis (Fig. 113*e*).

More often than not blanks during machining are located from two holes and a surface.

Solids of revolution have only two locations: an axial location which coincides with the axis of the solid and a height one that determines the dimensions along the axis (Fig. 113*f*). If there are axial locations, the casting and machining locations are made to

coincide, the common location being the axis of a hole selected to serve as a datum hole (half-shaded lozenge in Fig. 113*g* and *h*).

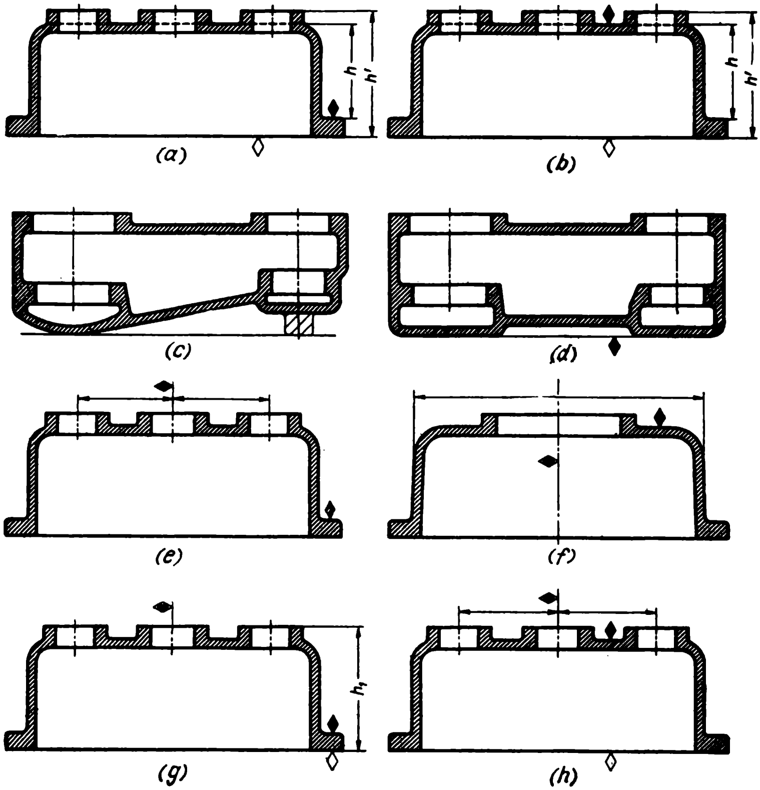


Fig. 113. Rough and machining locations

3.12. Variations in Casting Dimensions and Their Effect on the Design of Castings

Sand castings are liable to considerable dimensional variations which increase as the castings grow in size and complexity.

USSR State Standards ГОСТ 1855-55 and 2009-55 specify three grades of accuracy for the dimensions of grey-iron and steel castings. Figure 114*a-c* illustrates averaged values of permissible deviations for iron and steel sand mould castings, depending on their maximum overall size, for various distances from the casting location. Figure 114*d* shows the same dimensions for castings made from nonferrous alloys.

In sand mould casting with wooden patterns and cores moulded in wooden boxes the attainable accuracy does not exceed that of the

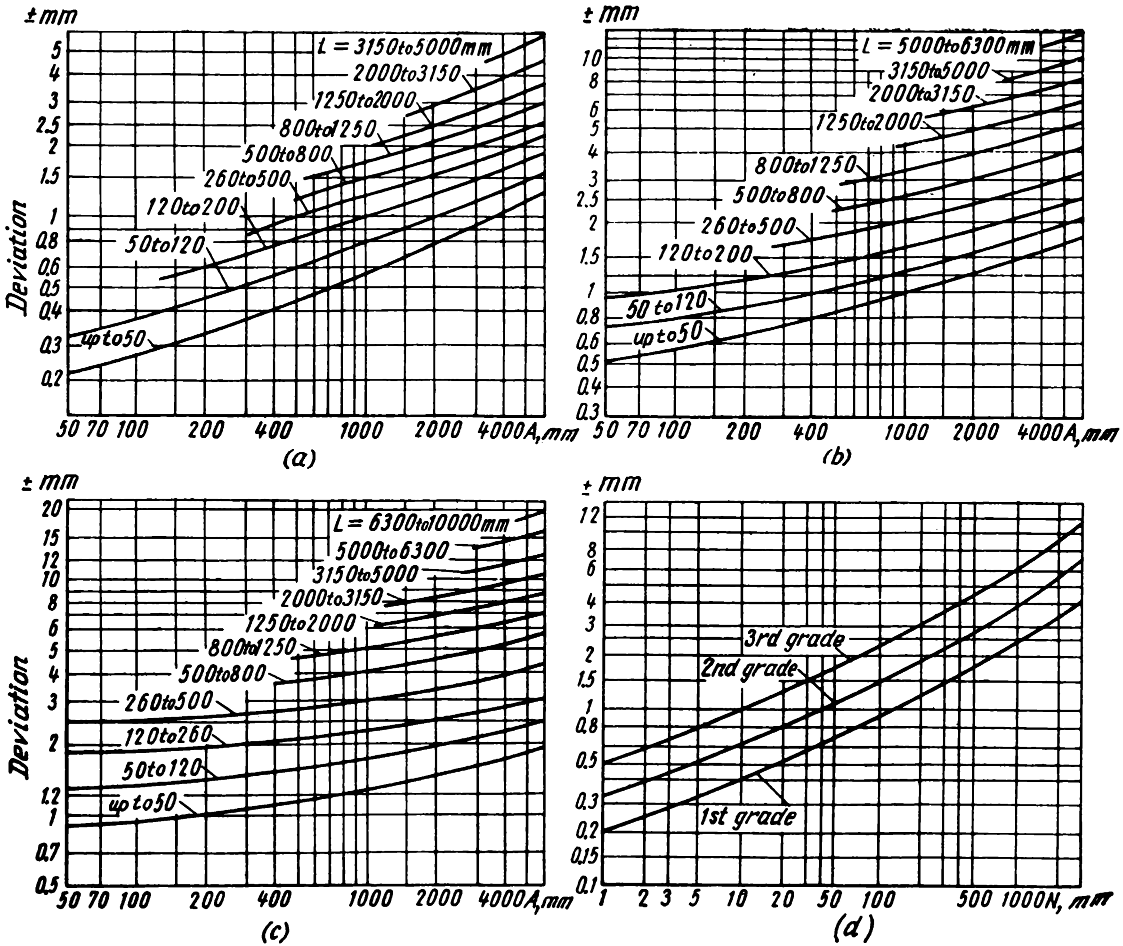


Fig. 114. Permissible deviations in dimensions versus the maximum overall size A of a casting

a—iron and steel castings, grade 1; *b*—iron and steel castings, grade 2; *c*—iron and steel castings, grade 3; *d*—castings from nonferrous alloys (N—nominal dimension)

3rd grade. Accuracy can be enhanced by using metal patterns and core boxes, mechanizing moulding, moulding in core and permanent moulds, and also by conducting the casting process properly.

The minimum deviations in the dimensions of a casting occur when moulding is done in one box. If a part is moulded in two or several boxes, there develop deviations because the boxes are displaced.

The top box (Fig. 115) can be displaced with respect to the bottom one by the amount of clearance a in the loose pins, which is attended by respective displacement of all the vertical surfaces moulded in the top box. This may greatly change the nominal thickness N of the walls.

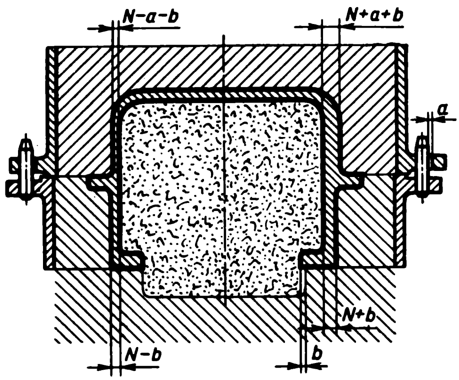


Fig. 115. Appearance of inaccuracies in casting in two moulding boxes

The surfaces moulded by cores may be shifted with respect to the surfaces moulded by the pattern due to inaccurate core installation in the mould (shift b in Fig. 115). Displacements reach their maximum in the top half mould where the shifts of the half moulds and the core are summed up.

In an unfavourable case (displacements of the core and half moulds are in opposite directions) the variations in the thickness of the vertical walls in the top half mould, equal to $\pm(a + b)$, exceed those in the lower half mould ($\pm b$) about twice.

Dimensional deviations of horizontal surfaces occur as a result of inaccurate installation of cores in the vertical direction, because of foreign matter getting on the parting planes of moulding boxes and cores, etc.

As a rule, the surfaces moulded in the bottom box are more accurate than the ones moulded in the top box. Surfaces moulded by the pattern are more accurate than those moulded by inner cores.

Among the other causes of inaccuracy are the deviations in the dimensions of the pattern set from the nominal sizes, the change in the dimensions of cores upon drying, cracking of the patterns in storage, the change in the dimensions of the mould occurring when rapping the patterns during extraction, variations in shrinkage due to the different pliability of cores, and warpage of the casting under the action of shrinkage stresses.

Variations in the dimensions of castings are reflected in the system of machining allowances according with the USSR State Standards ГОСТ 1855-55 (grey irons) and 2009-55 (steels). The amount of allowance depends on the accuracy grade and dimensions of

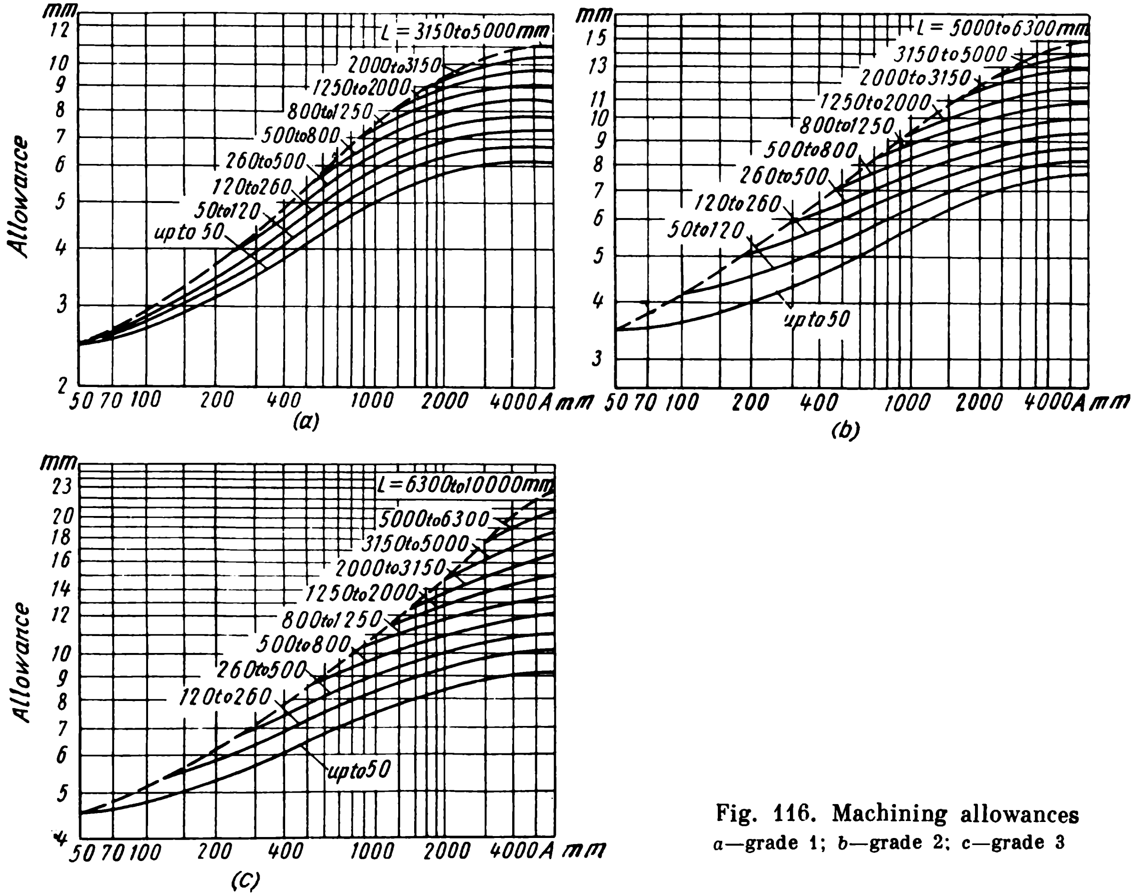


Fig. 116. Machining allowances
a—grade 1; *b*—grade 2; *c*—grade 3

castings, the nominal distance of the given surface from the location, the position of the surface in pouring (bottom, top, side) and the type of casting alloy.

Figure 116*a-c* illustrates averaged values of standard allowances for grey iron castings of various grades of accuracy, depending on the maximum overall dimension A of the casting for various distances L of the surface from the location.

The diagrams show allowances for upper surfaces of type m (Fig. 117) which have the maximum values because such surfaces are less accurate mainly due to the accumulation in the top stock of nonmetal inclusions, slag and other admixtures which are subject to removal in machining. The machining allowances for bottom surfaces n and side surfaces o are 20-30 per cent less than those for the top surfaces. Allowances for steel castings are 25-40 per cent greater than for iron castings.

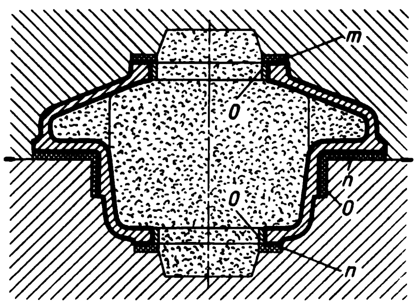


Fig. 117. Determining the amount of allowance

Variations in casting dimensions are especially important in sections where rough walls join work surfaces. Machining accuracy is many times higher than casting accuracy. A cast part may schematically be considered a rigid frame made up of work surfaces surrounded by a "floating" envelope of rough surfaces.

Let us denote the magnitude of possible displacements of rough surfaces by the letter k.

The following rules should be observed when designing castings:

(1) protruding work surfaces must lie above the adjacent rough surfaces by the amount k (Fig. 118*a*), this preventing the tool from cutting into the adjacent rough surfaces (Fig. 118*b*);

(2) sunk work surfaces must lie below the rough surfaces by the amount k (Fig. 118*c*) to allow a full reach by the tool (Fig. 118*d*) and prevent rough spots;

(3) the thickness of walls adjoining work surfaces (Fig. 118*e*) must be greater by the amount k than the thickness m required by the design. Otherwise, the walls may be impermissibly thinned should the cast surfaces be displaced (Fig. 118*f*).

Figure 119 shows how these rules are applied to hubs (Fig. 119*a*), bosses (Fig. 119*b* and *c*) and flanges (Fig. 119*d* and *e*).

Jointing planes should be connected with the nearest rough walls by surfaces perpendicular to the machining plane, the height of such surfaces being not less than k (Fig. 120), otherwise the contour of the joint may be distorted.

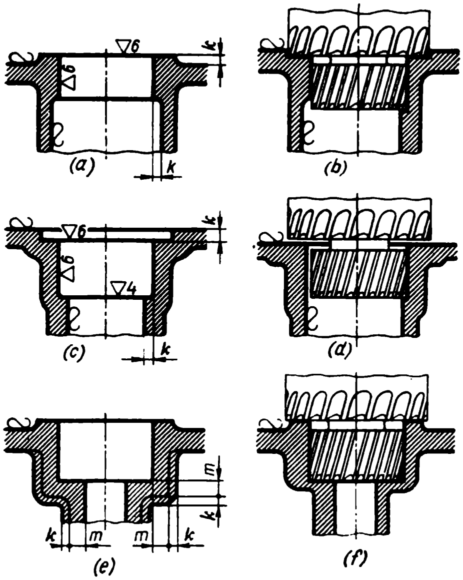


Fig. 118. Transitions between machined and rough surfaces

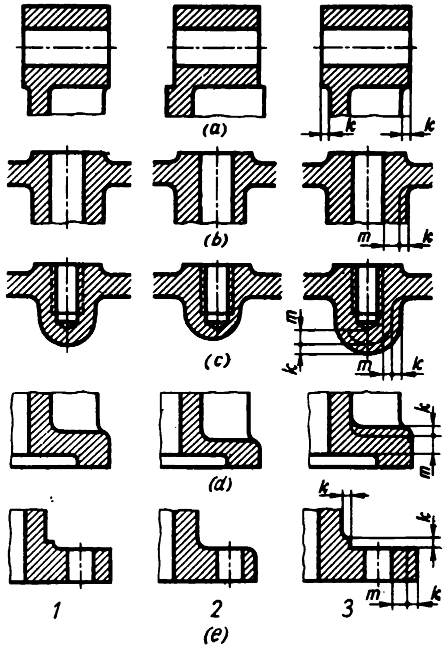


Fig. 119. Transitions between machined and rough surfaces

1—assigned shape; 2—shapes that can be obtained with casting errors; 3—shapes accounting for the displacement k of cast surfaces

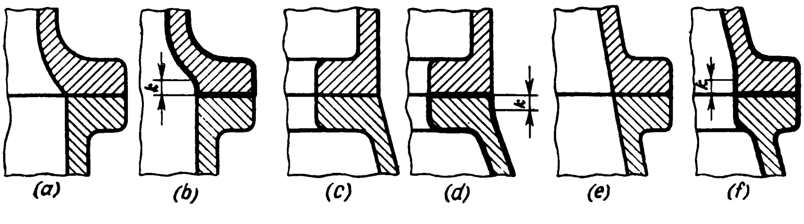


Fig. 120. Conjunction of jointing surfaces

a, *c*, *e*—wrong; *b*, *d*, *f*—correct

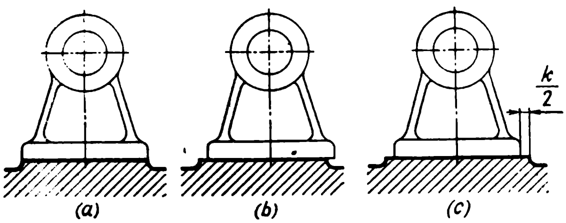


Fig. 121. Shape of joint surfaces

Mounting pads on housing-type components (Fig. 121*a*) should be designed with reserve *k* over the contour (Fig. 121*c*) so that the mounted part does not overhang (Fig. 121*b*).

The value of *k* depends on the accuracy and overall dimensions of the casting and the distance of a given element from the casting and

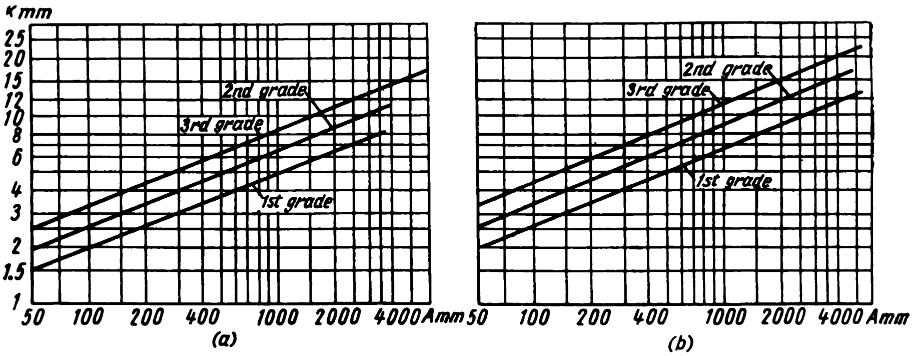


Fig. 122. Diagrams for determining the value of k (A is the maximum overall dimension of casting)
a—iron casting; *b*—steel casting

machining locations as well, and, in the general case, is determined by calculating dimensional chains. Practical designing, however, requires a simpler method.

The value of *k* can be found from the machining allowances (see Fig. 116) since the latter are determined by the same parameters as *k* (the maximum overall size of the casting, distance from the casting locations, grade of casting accuracy). To dispense with calculating the distances to the locations, the upper limits of allowances (dashed lines in Fig. 116) may be taken, which will go into the safety margin. Accounting for the fact that the diagrams give the maximum allowance values (for the top surfaces), a reduction factor of 0.7 should be introduced.

Figure 122*a*, *b* shows the values of *k* calculated by this method for iron and steel castings of the 2nd and 3rd grades of accuracy. The values of *k* may be directly utilized to find the required distance between rough and work surfaces.

The wall thickness of bosses may easily be found from the relation $S = as$ where s is the mean wall thickness of the casting and a is a coefficient equal to 1.5, 1.7 and 1.8 for the 1st, 2nd and 3rd grades of accuracy, respectively. These relations practically assure against excessive reduction in the wall thickness.

3.13. Dimensioning

The dimensions of cast parts on drawings must specify the position of casting and machining locations, and also account for size deviations.

The principal rules for the dimensioning of cast parts are as follows:

(1) rough surfaces must be related to a casting location either directly or by means of other dimensions;

(2) the initial machining location must be related to a casting location, all the remaining dimensions of the work surfaces being related to the machining location either directly or by means of other dimensions.

Casting dimensions must never be related to the dimensions of work surfaces or vice versa, except for the case when the casting and machining locations coincide (the case of axial locations).

These rules must be observed for all three coordinate axes of the casting.

Dimensioning of a cast part is illustrated in Fig. 123. Dimensioning according to Fig. 123*a* is wrong. The distance between the work surfaces, related

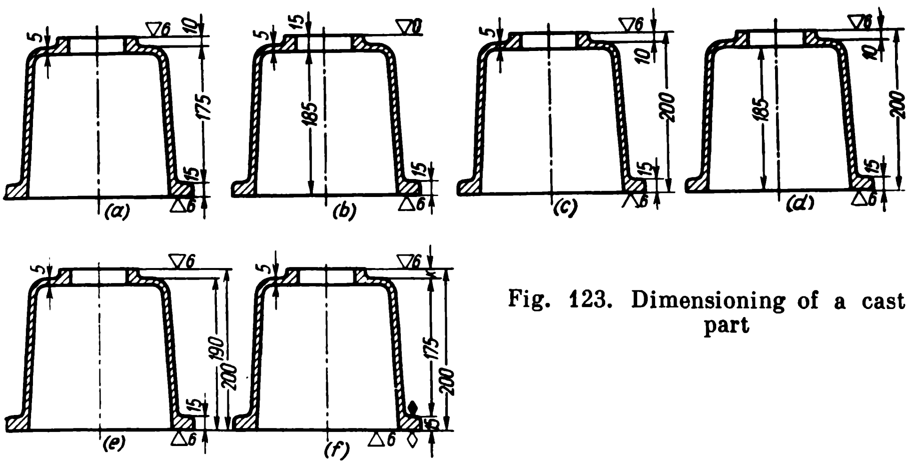


Fig. 123. Dimensioning of a cast part

to the rough surfaces by the sum of the dimensions 15, 175 and 10 mm, varies in this case within wide limits together with the variations in the dimensions of the rough surfaces.

The same error is made in the design in Fig. 123*b*, where the distance between the work surfaces is specified by the sum of the dimensions 185 and 15 mm.

When dimensioning is as in Fig. 123*c* the distance between the work surfaces (200 mm) is maintained within the necessary narrow limits (within the machining tolerance). The error lies in that the rough surfaces are related to the adjacent work surfaces (dimensions 15 and 10 mm). It is practically impossible to maintain this coordination. The position of the rough surfaces varies within the casting accuracy limits, entailing variations in the distance to the work surfaces.

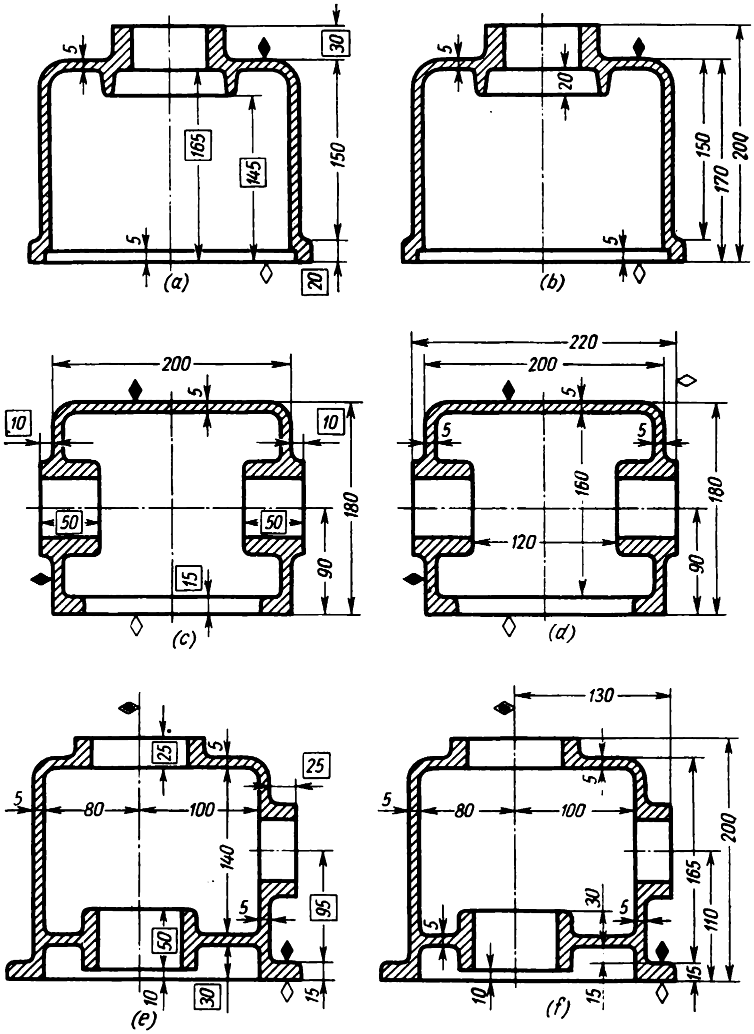


Fig. 124. Dimensioning of cast parts

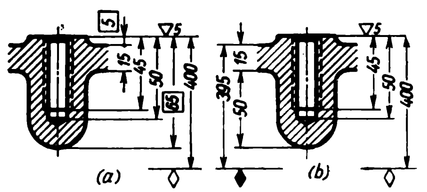


Fig. 125. Dimensioning of bosses

In Fig. 123*d* the error is aggravated by the fact that the thickness of the upper horizontal wall (directly specified in the previous cases by the dimension 5 mm) is determined by the height of the inner cavity, which is related to the lower work surface (dimension 185 mm). This introduces one more source of inaccuracy. The thickness of the wall will vary within wide limits.

In the dimensioning system according to Fig. 123*e* the position of the lower work surface is specified by two dimensions—one measured from the upper rough surface of the part (dimension 190 mm) and the other, from the upper rough surface of the flange (dimension 15 mm). It is practically impossible to preserve this coordination.

Figure 123*f* shows a correct system. The casting location is the top rough surface of the flange (blackened lozenge). The initial machining location (the bottom surface of the flange, marked by a light lozenge) is related to the casting location by the dimension 15 mm. The top work surface is related to the machining location (dimension 200 mm). The top rough surface is coordinated from the casting location (dimension 175 mm), and from it, the thickness of the upper wall (dimension 5 mm).

The distance *k* between the top work surface and the upper rough wall is the closing link in the dimensional chain and serves as a compensator for the deviations in the position of cast surfaces. Since the value of *k* is not specified on the drawing, it is not accounted for when checking the part. It stands to reason that the nominal value of *k* must be larger than the maximum possible displacement of the upper wall caused by casting inaccuracies.

Examples of wrong and correct dimensioning of cast parts are illustrated in Figs. 124 and 125 (wrong dimensions are given in squares).

Chapter 4

Design of Parts to Be Machined

Machining is one of the most laborious and expensive methods of manufacture and amounts to 70 per cent of the cost of a product.

The principal production methods of increasing the machining efficiency are as follows.

1. Reduction of *machining time* (intensification of cutting processes). These methods include *high-speed cutting* (increasing the main cutting speed), *heavy-duty cutting* (increasing the cutting feed and depth), and high-productivity processing (machining with multipoint tools; internal and external broaching; turn-milling; etc.).

2. Reduction of *handling time* (the use of quick-acting appliances; automatic feed, mounting, fastening and removal of the blank; machining to preset operations; automatic readjustment of the machine set-up; and automatic control). Another form of this method is the consecutive machining of blanks in multistation fixtures.

3. Matching of process operations in time (proper sequencing of operation elements). This method includes machining with combination tools and multiple-tool machining (multi-cutter turning and planing; milling with a set of milling cutters). The method is most fully embodied in unit-head machine tools in which several surfaces of a blank are machined simultaneously.

4. Simultaneous machining of several blanks (parallel and parallel-consecutive machining in multi-station fixtures; continuous machining on rotary and drum-type machine tools and on vertical turret lathes).

5. Rapid transfer of blanks from machine to machine (mechanical transportation of blanks; rational arrangement of equipment). Automatic and semi-automatic transfer lines, especially those of rotary type, are most productive.

Mass and stable production and all-round unification of designs with few models are requisite for the application of highly productive machining methods, special manufacturing riggings and special-purpose machine tools.

When designing parts to be machined the labour required by the machining process should be reduced to the minimum, and high

quality, reliability and durability of machines ensured at the same time.

When designing parts for machining the following rules are to be observed:

reduce the length of work surfaces to the design minimum required;

decrease the machining allowances to the minimum;

manufacture parts by the most productive methods which do not involve chipping (forging, cold upsetting, coining, etc.);

widely use shape steel rolled stock, leaving most of the surfaces in the as-rolled condition;

make parts from blanks having their shape as close as possible to that of the final product;

use composite structures to make easier the manufacture of labour-consuming parts;

avoid unnecessary precise machining. In each particular case use the lowest grade of accuracy ensuring proper functioning of the unit and meeting interchangeability requirements;

provide for the use of the most effective machining methods (with calibrated multipoint tools, etc.);

provide as far as possible for through-pass machining, which is the principal condition for increasing productivity and obtaining high-accuracy and finish standards of the machined surfaces;

if through-pass machining proves impossible, ensure that the tool overtravel is sufficient to obtain well-finished and accurate surfaces;

ensure convenient approach of the cutting tool to the work surfaces;

make it possible to machine the maximum number of surfaces during one operation on one machine in a single setting and with one and the same tool;

shape parts of repeated and mass application so as to make them suitable for group machining with the use of combination tools;

provide for the machining of accurate coaxial and parallel holes in a single setting to obtain good alignment and precise centre distances;

assure a clear distinction between the surfaces machined in different operations, by different tools and to different accuracies;

provide for the distances between the work and nearest rough surfaces which will make machining possible with the maximum variations in the blank size;

avoid joint machining of assembled parts, which disturbs the continuity of the production process, impairs interchangeability and makes it difficult to replace parts during operation;

reduce the range of the tools employed by unifying the size and shape of the elements to be machined;

in piece and small-lot production reduce the number of special cutting tools to a minimum, using standard tools as far as possible;

impart to the work surfaces such a shape as will make the tool operate smoothly without impacts;

relieve cylindrical multipoint tools (drills, reamers, counterbores, etc.) from a unilateral pressure in operation;

impart to the portions to be machined a high and uniform rigidity ensuring an accurate machining to good finish and making for the use of efficient processing methods;

provide convenient datum surfaces for size control with the use as far as possible of universal measuring tools.

4.1. Cutting Down the Amount of Machining

The examples in Fig. 126 show how superfluous machining can be eliminated. In the fastening unit of a guideway (Fig. 126*a*) it is

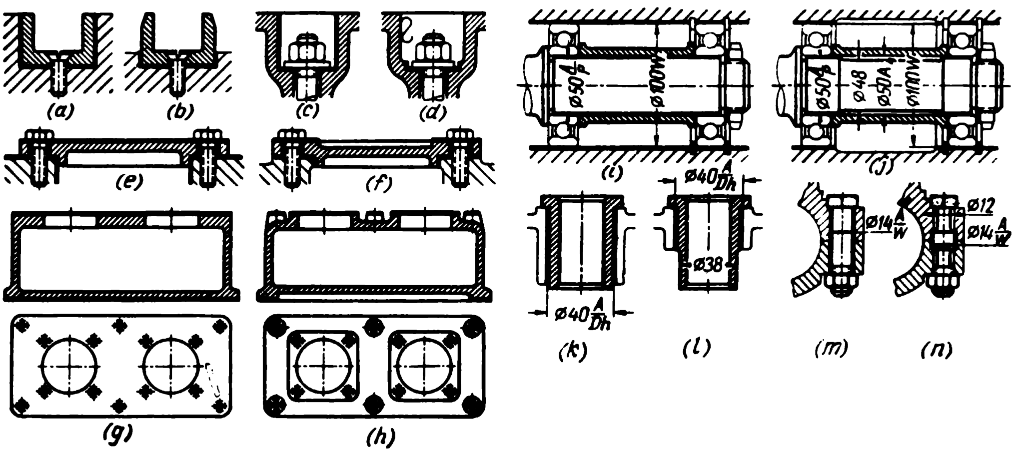


Fig. 126. Cutting down the amount of machining

advisable to reduce the depth of the locating slot in the housing (Fig. 126*b*) to an amount sufficient for reliable locking.

In cast parts (pit for a fastening bolt, Fig. 126*c* and *d*; cover, Fig. 126*e* and *f*; housing, Fig. 126*g* and *h*) the surfaces to be machined should be arranged above the adjacent rough surfaces.

In an antifriction bearing unit (Fig. 126*i*) precision machining should be applied to strictly limited portions of the working surfaces (Fig. 126*j*).

Figure 126*k* and *l* shows the shortening of the press-fitted portion of a bushing in a housing, and Fig. 126*m* and *n*, the reduction of the centring portion of a dowel bolt.

For parts made of round rolled stock the labour required for machining and the amount of chips removed can be reduced mainly by decreasing the difference between the diameters of the parts, espe-

cially the largest diameters which determine in the first place the amount of the cut-off material.

The shoulder on a stepped shaft (Fig. 127*a*) increases the diameter D of the blank and sharply increases the amount of the cut-off metal. The large difference between the step diameters requires in turn more machining. The volume of the cut-off metal amounts to 135 per

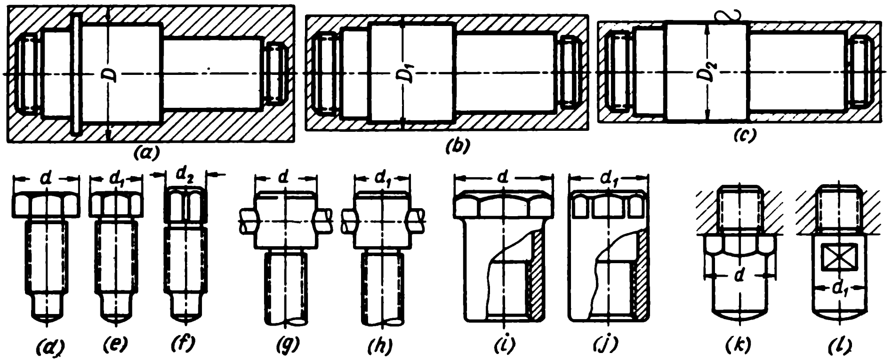


Fig. 127. Parts made of round rolled stock

cent of that of the final product. The coefficient of utilization of the material of the blank is 0.43, i.e., more than half of the blank metal is rejected as chips.

In the shaft design without shoulder and with a smaller difference in the step diameters (Fig. 127*b*) three times less metal is removed as compared with the previous case, thanks to the smaller diameter D. Most of this reduction to diameter D_1 (80 per cent) is due to the elimination of the shoulder. The coefficient of utilization of the material is increased to 0.7.

Figure 127*c* illustrates a further reduction in the amount of the metal removed, made on account of the part being manufactured from a cold-drawn bar with a diameter equal to the maximum diameter D_2 of the shaft. In this case the coefficient of utilization of the material is increased to 0.8.

Examples of cutting down the amount of machining by reducing the maximum diameter of parts are illustrated in Fig. 127*d-f* (pressure screw), Fig. 127*g, h* (tommy bar head), Fig. 127*i, j* (cap) and Fig. 127*k, l* (leg).

The diameter of a product should correspond to the standard diameters of round rolled stock. The maximum diameter of a product should be less than the nearest standard diameter of the bar by an amount equal to the diametral machining allowance a.

The value of a may be found from the ratio

$$a = b\sqrt[6]{DL}$$

where D = diameter of the surface to be machined, mm
L = length of the blank, mm
b = coefficient equal in various types of machining to:

Operation	Machining		Total allowance
	rough	finish	
Turning	0.5	0.4	0.9
Grinding	0.2	0.1	0.3

It is better to make mass-produced fasteners from sized rolled stock, leaving the maximum possible portion of the blank surface unmachined.

Figure 128*a* and *b* shows how labour input can be reduced by making a stud from a cold-drawn sized bar.

The design of a hexagon nut with an annular collar (Fig. 128*c*) is unsuitable for mass production. Such nuts can be manufactured only by the piece. The nuts in Fig. 128*d* and *e* are made of hexagon bar steel.

A cylindrical serrated nut in which the serrations come onto the surface of the cylinder (Fig. 128*f*) is unsuitable for mass production because such nuts have to be milled individually. Correctly designed nuts which can be manufactured from cold-drawn sized bar are shown in Fig. 128*g* and *h*.

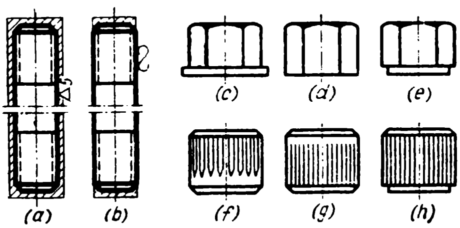


Fig. 128. Mass-produced fasteners made of rolled stock

Much less machining will be required if pipes are used to make hollow cylindrical parts.

Figure 129*a* presents a hollow pillar made from solid bar steel. The amount of machining will be less if the pillar is made of seamless pipe and the internal surface left rough (Fig. 129*b*), and still less, if the collar diameter is reduced (Fig. 129*c*).

Figure 129*d* shows the shell of an antifriction bearing. It takes much labour to make part *1* (Fig. 129*e*) from a cylindrical blanks; besides, 85 per cent of the blank volume is wasted in chips.

In Fig. 129*f* the shell is divided into three parts. The side cheeks are made of plate steel and the middle portion, of thin-walled pipe. In mass production part *1* is preferably forged (Fig. 129*g*).

When making machine parts by slitting cylindrical blanks (Fig. 129*h-k*), the angular dimensions of the parts should be assigned

so as to fit an integral number of them within the blank circumference, with due regard for the slitting cutter thickness, thus making the maximum use of the blank.

The parts in Fig. 129*h* and *j* are designed without allowance for the slitting cutter thickness *s*. Therefore in the first case about half

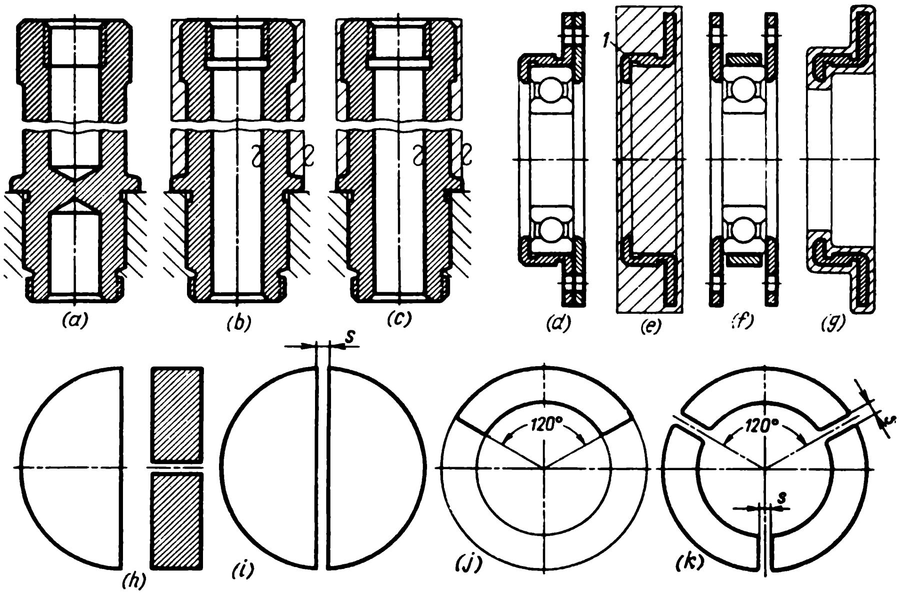


Fig. 129. Manufacture of cylindrical parts

and in the second, one third of the blank is wasted. In Fig. 129*i* and *k* the dimensions of the parts are selected with due regard for the slitting, and the entire blank is completely utilized.

4.2. Press Forging and Forming

It is most advisable to make parts from blanks having their shape close to that of the final product, obtained by hot forging in closed-impression dies. This reduces the amount of machining and increases the strength of the part, thanks to the compaction of the metal, formation of fibre structure and fine equiaxial grains resulting from recrystallization which occurs as the blank cools down.

All other conditions being equal, forgings are stronger and lighter, and require less machining than composite parts.

The use of dies is economically justifiable in mass production where the initial investments in the manufacture of dies are rapidly recouped because of increased output and reduced machining. However, thanks to the high strength of forged parts, the method is often used in the manufacture of important machines irrespective of the scale of output and manufacturing costs.

The highest accuracy and surface finish standards are provided by cold sizing (coining) applied as a final operation after hot forging. In some cases coining completely dispenses with machining.

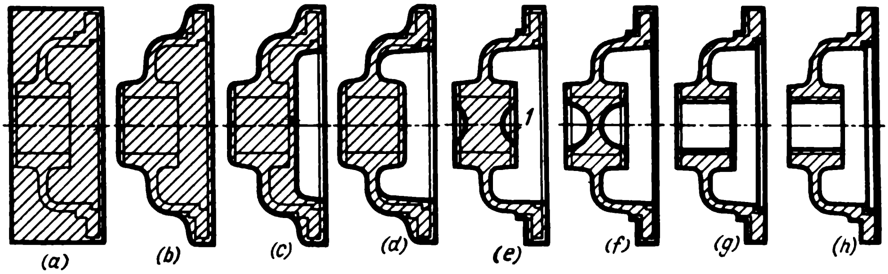


Fig. 130. Methods of making a cup-shaped part

Figure 130 illustrates methods of making a cup-shaped part (shown on the drawing by thin lines).

Much labour is required to turn the part out of a cylindrical blank (Fig. 130*a*). Besides, the part is weakened because the metal fibres are cut.

Figure 130*b* shows a blank obtained by hammer forging in open dies with a profiled lower die and flat upper die; Fig. 130*c* and *d*

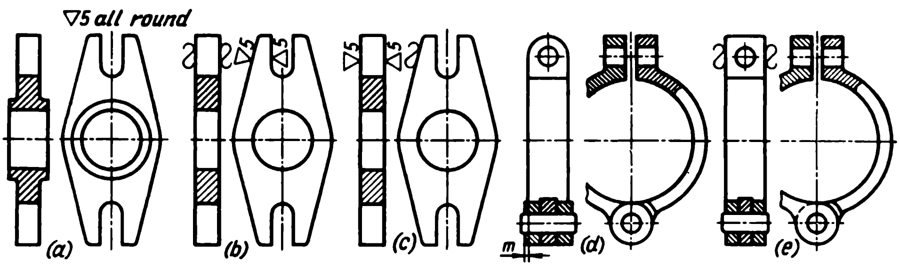


Fig. 131. Simplified machining of shaped parts

illustrates the same blank made with the use of profiled lower and upper dies.

When the blank is forged in a closed single-impression die (Fig. 130*e*) most of the surfaces take the final shape except for the surfaces to be machined. The hole is marked by recesses *1*. The flash in the hole is removed by machining or subsequent forging operations.

Forging in a finish impression (Fig. 130*f*) provides a higher wall accuracy and in this case smaller machining allowances can be assigned. The partition in the hole is cut out by a punching die.

A blank with pierced hole obtained on a horizontal forging machine is presented in Fig. 130*g*.

Cold sizing (coining) imparts the final shape to all surfaces (Fig. 130*h*) except for the surfaces which require a most precise machining (seating hole, centring recess, end face of flange).

Flat shaped parts are advisably made of plate material.

The laborious contour machining of the part shown in Fig. 131*a* can be simplified by making the parts of plate material (Fig. 131*b*) with gang form milling or shaping of the external contour. The required section can also be obtained by extrusion. The parts in this case are produced by cutting the extruded section to the required length (Fig. 131*c*).

The clamp shown in Fig. 131*d* requires arduous contour machining or die forging followed by all-round trimming. If the design is slightly changed (removal of lug projections *m*) such clamps can be made from plate (Fig. 131*e*) with form milling of the external contour.

4.3. Composite Structures

Composite structures are used in small-scale production when the manufacture of dies is economically unjustifiable.

Some examples of dismembering parts as a means of reducing the amount of metal wasted in chips are illustrated in Fig. 132, *1*, *2* (plug cock), *3*, *4* (piston) and *5-7* (pillar fastening). Parts are often dismembered to reduce the labour required for machining.

In the unit comprising a labyrinth seal and a self-expanding ring seal (Fig. 132, *8*) it is practically impossible to make part *a* because the cutting tool cannot approach the crests of the inner labyrinth and the spring-ring grooves. When the part is separated into two elements (Fig. 132, *9*) it can easily be machined.

Figure 132, *10*, *11* shows the simplification of the machining of an annular T-shaped slot by separating the part into two elements.

The part with an internal hub (Fig. 132, *12*) can be machined to the required accuracy only with the aid of a cup-shaped grinding wheel (Fig. 132, *13*). With the composite design (Fig. 132, *14*) the detachable hub is ground externally.

Figure 132, *15-34* shows examples of separating intricately shaped parts—pipe union (Fig. 132, *15*, *16*), cup-shaped part with an internal spherical surface (Fig. 132, *17*, *18*) and hollow shaft with an internal partition (Fig. 132, *19*, *20*).

It is difficult to machine cylindrical and spherical projections whose axes do not coincide with the rotation axis of the part. They can be machined on lathes only with the aid of special attachments (offset centre fixtures) and ground by means of cup-shaped wheels. Such parts are preferably made detachable.

The design of the carrier with rings made integral with the carrier housing (Fig. 132, *21*) is not sound. It is more practical to fit the

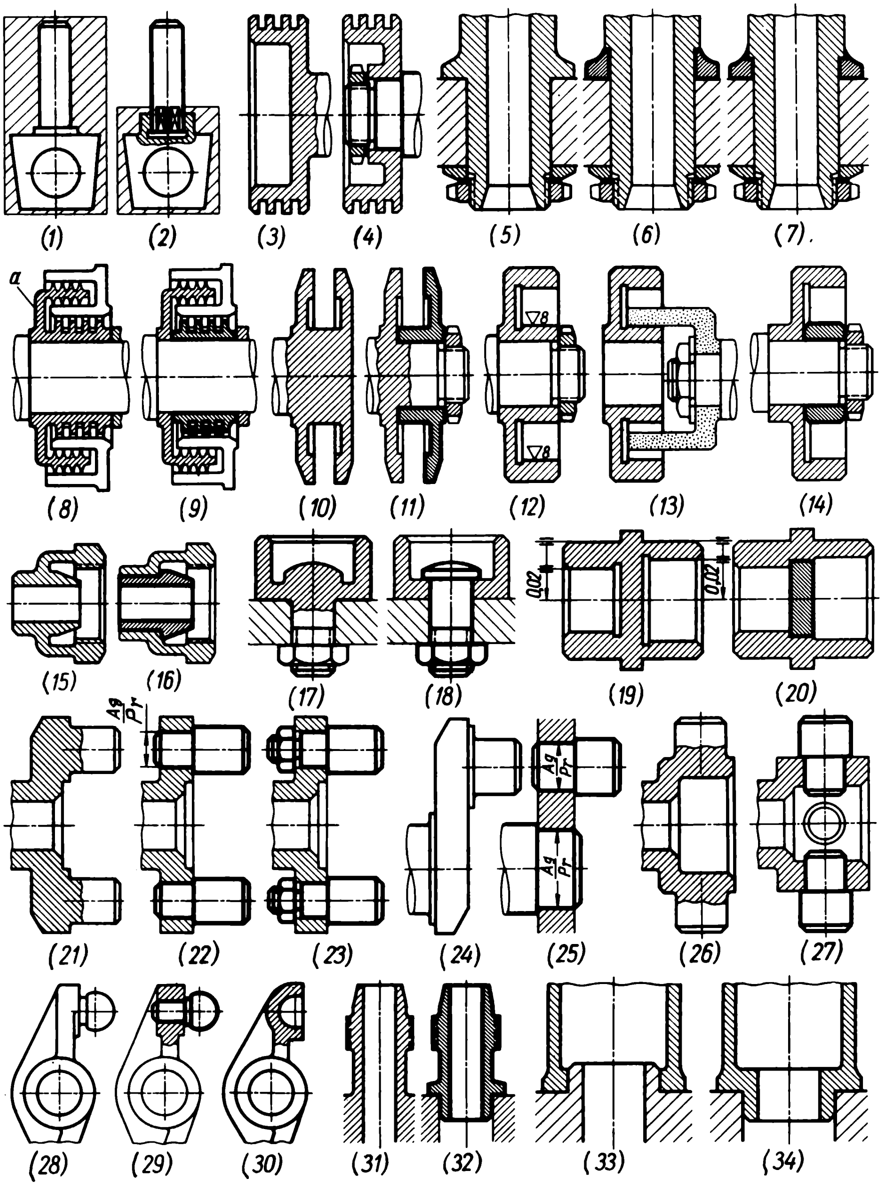


Fig. 132. Composite structures

pins in holes (Fig. 132, *22*, *23*) which can be accurately manufactured and coordinated with ease.

Projections may be made integral with the part if there are not more than two projections arranged on different sides of the part

(for example, frontal cranks, Fig. 132, *24*). The composite structure (Fig. 132, *25*) is however more practical although it is inferior in strength to the integral one.

Other examples of composite structures are presented in Fig. 132, *26*, *27* (cross-shaped carrier) and *28*, *29* (lever with a spherical striker). In the latter case, another design (Fig. 132, *30*) wherein the striker head is replaced by a spherical cup is just as valid.

External threads on the projecting members of housing-type components (Fig. 132, *31*) have to be cut manually. This is unsuitable for mass production, and it will be more practicable to make these parts detachable (Fig. 132, *32*).

Centring from external shoulders on housings (Fig. 132, *33*) should preferably be changed to centring from holes (Fig. 132, *34*).

4.4. Elimination of Superfluously Accurate Machining

Close-tolerance dimensions should be applied only when absolutely necessary. One should always select the lowest grade of accuracy permissible from the standpoint of interchangeability of parts and reliable operation of the given unit.

Surfaces whose manufacturing accuracy does not affect the operation of the unit as a whole should be made to lower grades of accuracy than the working surfaces.

Figure 133*a* shows a shaft mounted in rolling-contact bearings. The seating surfaces for the bearings conform to the 2nd grade of accuracy. The centring surfaces of intermediate bushings *1*, *2* and *3* and of grooved seal body *4* are machined to the same accuracy although rougher tolerances (to the 3rd or 4th grade of accuracy, Fig. 133*b*) can safely be assigned for these surfaces.

It is not necessary to assign close-tolerance dimensions for the internal diameter of the seal body *4* and for the external diameter of bushing *3* because a radial clearance of 0.5 mm exists between these surfaces. These dimensions may be given without tolerances.

When a ball bearing is locked with rings on the shaft and in the housing (Fig. 133*c*, *d*) it is not necessary to install the locking rings into grooves by a slide fit and machine them to the 2nd grade of accuracy since the fit of the rings and the accuracy of the bearing location are determined only by the overall dimension 24 A_3 between the extreme end faces of the grooves and the total thickness of the parts included in this interval (locking rings, bearing race). In order to simplify machining it is advisable to fit the locking rings into the grooves with an axial clearance of about 0.3 mm (Fig. 133*d*).

Figure 133*e* shows axial locking of a ball bearing in the housing by means of checks *5*. To ensure clearance-free installation the end faces of the housing are machined to the 2nd grade of accuracy to a dimen-

sion equal to the width of the outer bearing race (20S). The manufacture of the unit can be simplified by machining the end faces of the housing without keeping to close tolerances, the clearance-free installation of the bearing being ensured by means of a sized ring *6*

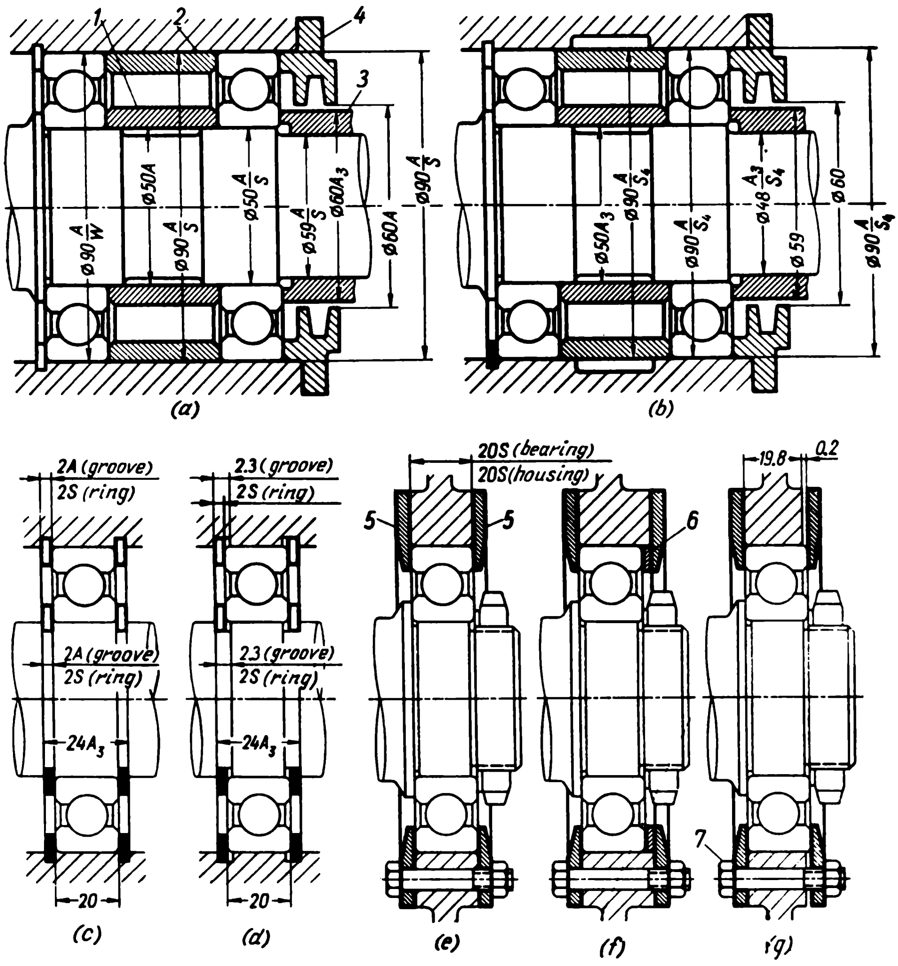


Fig. 133. Elimination of excessively accurate machining

(Fig. 133f). Another, and much simpler method is to make the housing wall thickness 0.1-0.2 mm smaller than the width of the bearing (dimension 19.8 in Fig. 133g). When fastening bolts *7* are tightened up the cheeks are elastically deformed and securely lock the bearing axially.

4.5. Through-Pass Machining

The through-pass machining where the cutting tool freely approaches and leaves the work surface is of great value for raising productivity and improving surface finish and accuracy.

The housing design in Fig. 134*a* is not good since the traverse of the cutting tool (face milling cutter) along the work surface is limited by the housing walls.

The cutting conditions vary with different portions of the surface. At first the blank is brought to the cutter axially, then the cutter

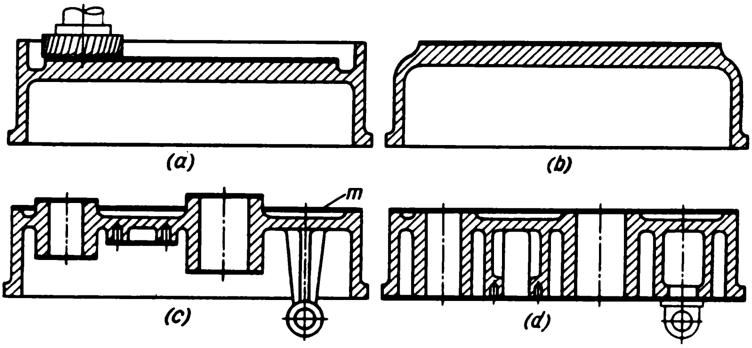


Fig. 134. Through-pass machining of frames

bites into the metal, in which case it is difficult to obtain fine surface. Several cuts are required to obtain more or less identical finish over the entire length of the machined surface.

Such productive methods as high-speed cutting, machining to preset operations and also gang machining cannot be applied in this case. Each workpiece has to be machined individually, much time being wasted to feed the milling cutter in and back it out, and adjust the setup to size.

In the correct design with the protruding work surface (Fig. 134*b*) the milling cutter operates with through feed and cuts the surface to the same finish at a high productivity.

Figure 134*c* shows a plate design unsuitable for mass production. The work surfaces are arranged at different levels, each surface requiring individual machining. Due to the presence of internal bosses, the contour of the upper flange *m* has to be machined with a combined cross and longitudinal feed of the work. The bracket with a transverse hole, which protrudes below the lower surface of the plate, makes it difficult to machine this surface and mount the plate properly when machining the upper surfaces. It is inconvenient to drill the transverse hole in the bracket, especially if the hole is far from the external edges of the plate.

In the good design (Fig. 134*d*) all the work surfaces are brought to the same level. The bracket is made detachable. The machining is done in two stages: first the upper surface is cut and then the lower one.

Figure 135 shows examples of machining accurate holes. In design *1* the bearing is installed in a split housing (radial assembly), in a recess limited on both sides by walls. It is extremely difficult to machine the seating surface of the recess.

Design *2* of axial assembly (a bearing mounted in a solid housing) is likewise unsuitable. Accurate machining of the seating surface is hampered by the shoulder that locks the bearing in the axial direction.

The designs where the seating surface is through-pass machined are correct. In this case the bearing is secured axially by locking rings (design *3*) or intermediate bushings (design *4*) one of which is fastened in the housing and the other serves to tighten up the bearing race. Figure 135, *5*, *6* illustrates irrational (*5*) and rational (*6*) mounting of a rolling-contact bearing.

The mounting of rolling-contact bearings in a gear with a collar used to lock the bearings (design *7*) is unsuitable. In this case it is especially difficult to ensure the concentricity of the seating surfaces machined in different settings. When the collar is replaced with a locking ring (design *8*) the hole can be through-pass machined.

If a ram is fitted into a blind hole (design *9*), it is difficult to machine the hole and lap-in the ram. In this case a through hole is required (design *10*).

In a cover with a shaped flange *m* machined by milling (design *11*) it is better to make the flange of such a shape as will allow through-pass machining (design *12*).

In design *13* the nut-seating surfaces are face milled individually. Changing the shape of the seating surfaces (design *14*) makes it possible to machine them all in one go, thereby appreciably increasing the efficiency of machining.

Slots (design *15*) should preferably be made open (design *16*) as in this case their machining is simplified and their side faces can be made more accurately.

Some changes in design that allow through-pass machining are illustrated in Fig. 135, *17*, *18* (fitting a bushing into a housing); *19*, *20* (torque-unit transmitting in a flanged connection); and *21*, *22* (fastening a shaft by means of a pin).

Figure 135, *23*, *25* shows wrong designs of housings with holes arranged in line. If there are solid walls the holes must be machined with an end cutting boring bar whose end is unstable and deflects under the effect of the cutting force.

In Fig. 135, *24*, *26* the housings are provided with holes for passing the boring bar and in this case the end of the bar can be steadied with a rest.

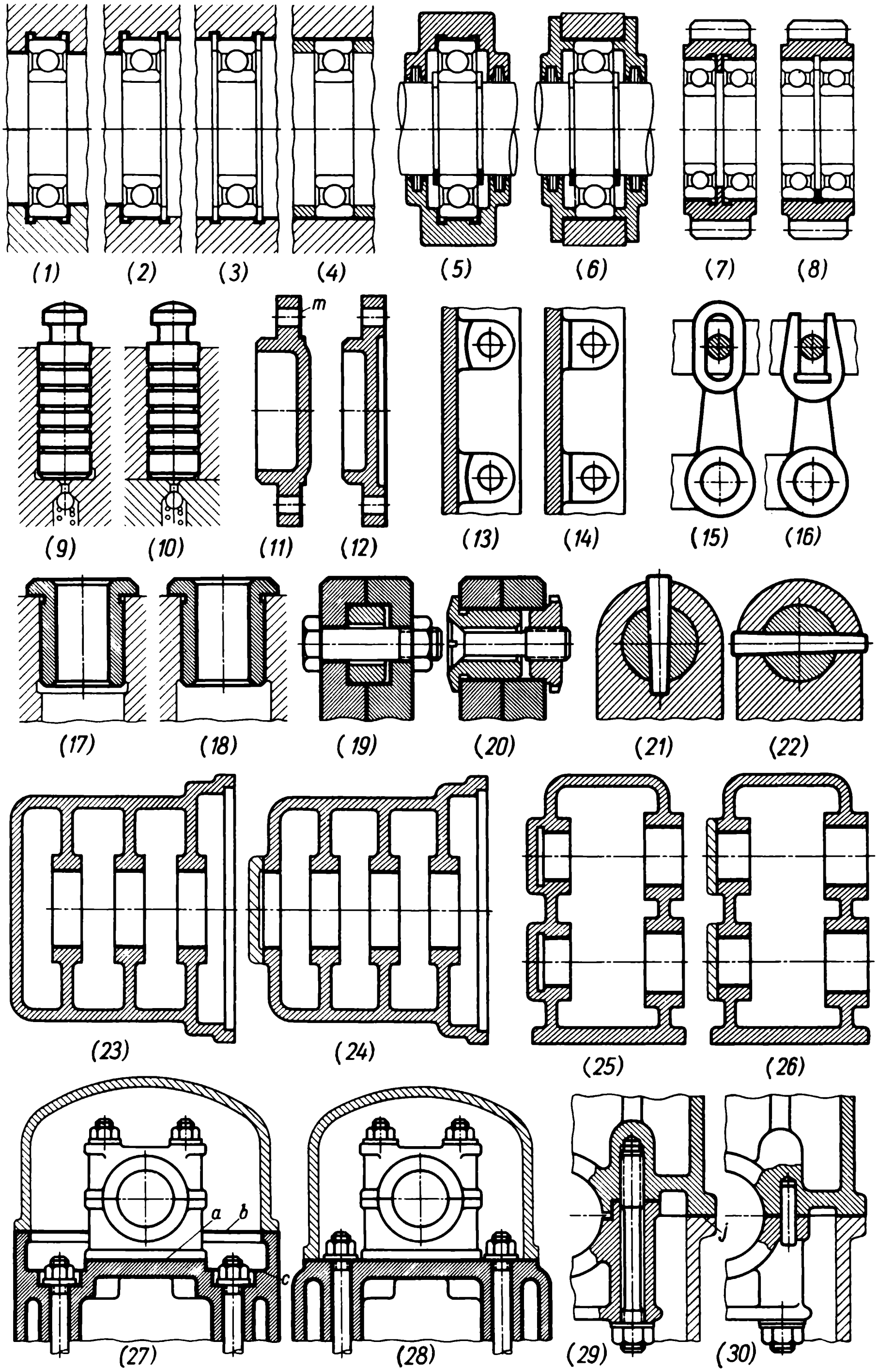


Fig. 135. Through-pass machining of holes

Figure 135, *27*, *30* shows how machining can be simplified by arranging the work surfaces in one plane. In the design of an engine head (Fig. 135, *27*) the machining is done on plane *b* where the head joins the cover, on plane *a* where the camshaft bearings are mounted, and on the seating surfaces of the fastening nuts.

A good design is the one in which all the three surfaces are brought to the same level and machined in one go (Fig. 135, *28*).

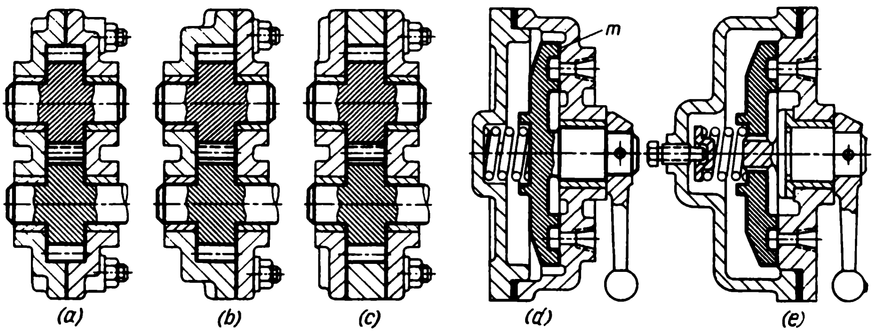


Fig. 136. Through-pass machining of holes and surfaces

In the crankcase bearing unit (Fig. 135, *29*) the bearing cap is located by means of shoulders, which prevents the through-pass machining of the jointing surfaces of the crankcase and bearing.

In the design shown in Fig. 135, *30*, the cap is located with set pins, and the through-pass machining of the surfaces is thus made possible.

The design of the gear pump in Fig. 136*a* is unsuitable for mass production. The seats for the gears are blind and are arranged in different halves of the housing. In such conditions it is difficult to coaxially align the seats. A better design is presented in Fig. 136*b* where the seats are situated in one half of the housing. The best design is the one where the housing is composed of three parts (Fig. 136*c*). The seats in the middle portion of the housing and the working surfaces of the housing cheeks are through-pass machined.

Figure 136*d* shows an irrational design of a flat slide valve. The working surface *m* of the housing is in a cylindrical recess, and it is impossible to grind this surface to the required accuracy. The conditions of grinding the working surface of the slide valve are likewise unfavourable. Even a slight out-of-squareness of the surface with respect to the valve axis may disturb the tightness of the seal.

In the design shown in Fig. 136*e*, the working surfaces of the housing and the slide valve can be through-pass machined on a surface-grinding machine.

This design also incorporates other improvements. The slide valve is connected with the shaft by splines which makes for an easy self-alignment of the slide valve with respect to the housing and increases the reliability of the seal. The spring that presses the slide valve rests against the cover of the housing via a spherical joint. This uniformly distributes the pressure force acting on the slide valve and reduces friction when the slide valve rotates.

4.6. Overtravel of Cutting Tools

Sometimes through-pass machining is impossible for design considerations. In such cases provision should be made for an *overtravel* of the cutting tool with respect to the work surface to a distance sufficient to obtain the specified finish and accuracy.

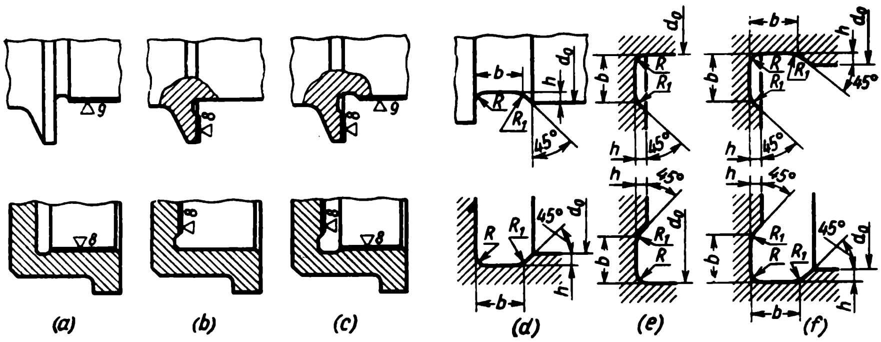


Fig. 137. Grooves for the overtravel of cutting tools

When machining accurate stepped cylindrical surfaces the overtravel of the tool is ensured by means of grooves several tenths of a millimetre deep cut at the section transitions.

If a cylindrical surface alone is subject to precision machining, use is made of cylindrical recesses (Fig. 137*a*), and when end faces are to be accurately machined (Fig. 137*b*) end recesses are cut. Diagonal grooves are made when a cylinder and the adjoining end face are to be precision machined (Fig. 137*c*). The shapes of grooves for the overtravel of a grinding wheel are illustrated in Fig. 137*d* (cylindrical grinding), *e* (face grinding) and *f* (cylindrical and face grinding).

The dimensions of the grooves (in mm), depending on the diameter d_0 of the cylinder, are given below.

d_0	up to 10	10-50	50-100	over 100
b	2	3	5	8
h	0.25	0.25	0.5	0.5
R	0.5	1.0	1.5	2.0
R_1	—	≈ 2 h	—	—

Figure 138 presents the shapes of adjoining surfaces of standard parts used in mechanical engineering.

It is impossible to finish machine the portion of a stepped shaft (Fig. 138, *1*) where the cylindrical surface adjoins the end face of the

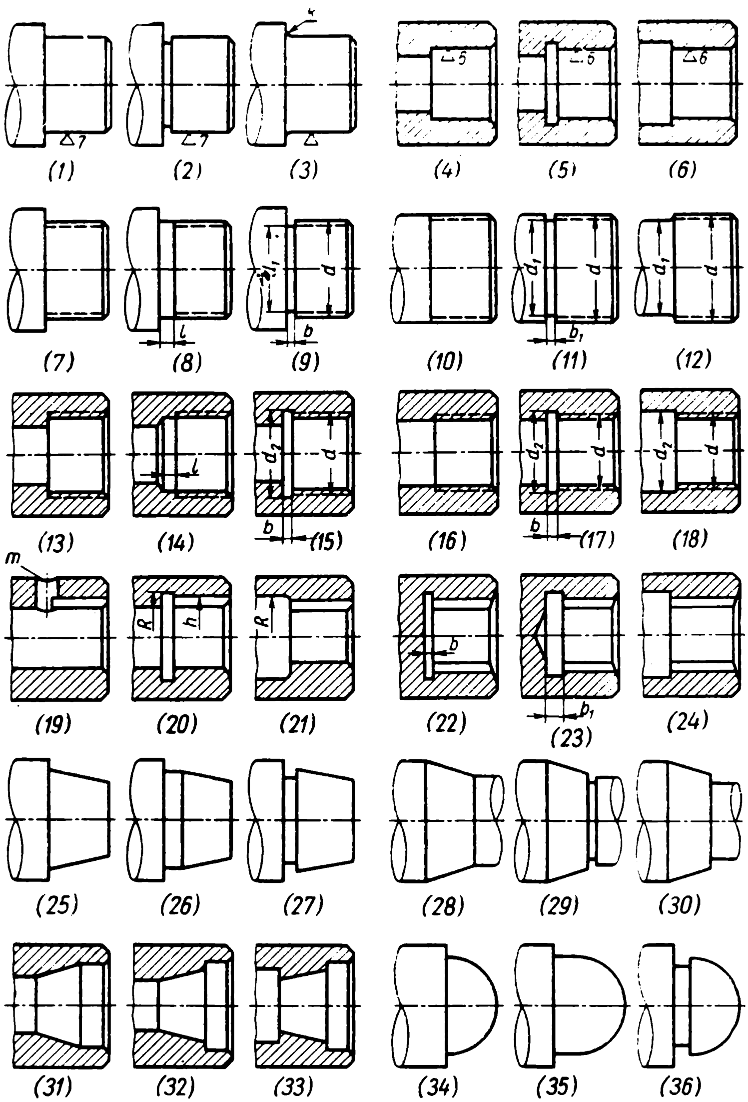


Fig. 138. Adjoining surfaces

collar. To ensure tool overtravel, a groove should be provided at the point of transition (Fig. 138, *2*). This method is not recommended for heavily loaded parts because recesses act as stress concentrators.

In such cases a filleted transition (Fig. 138, *3*) is required, made with a round-nose tool in turning, and with a round-face wheel in grinding.

To obtain accurate inner surfaces (Fig. 138, *4*), it is necessary to introduce undercut grooves (Fig. 138, *5*) or, better still, to ensure through-pass machining (Fig. 138, *6*).

Designs in which threads on cylindrical stepped portions are cut close to the end faces of the steps (Fig. 138, *7*, *13*) are practically impossible. Threads should terminate at a distance $l \geqslant 4S$ from shoulders or end faces (Fig. 138, *8*, *14*), where S is the thread pitch, or separated from the adjacent surfaces by a groove (Fig. 138, *9*, *15*) with a diameter $d_1 \leqslant d - 1.5S$ for external threads and $d_2 \geqslant d + 0.25$ for internal threads, where d is the nominal thread diameter in mm.

When cutting external threads with threading tools or dies the width of the grooves is, on the average, $b = 2S$, and when cutting internal threads, $b = 3S$. It is advisable to observe this rule also in the case of smooth shafts (Fig. 138, *10*, *11*) and holes (Fig. 138, *16*, *17*).

Surfaces adjacent to threads should preferably be arranged lower (Fig. 138, *12*, *18*) to allow through-pass machining. The diameters d_1 and d_2 of such surfaces are determined from the relations iven above.

When cutting longitudinal slots in holes, provision should be made for the slotting tool exit, for example, into a transverse bore m (Fig. 138, *19*) or into an annular groove (Fig. 138, *20*) of radius $R \geqslant \sqrt{h^2 + \frac{c^2}{4}}$ (where h is the distance from the slot bottom to the centre and c, the slot width). It is better for the adjacent surface to be located below the slot bottom (Fig. 138, *21*).

The design of a blind hole with splines machined by broaching (Fig. 138, *22*) is wrong: the width b of the groove beyond the splines is not enough for the overtravel of the broaching tool. In the design shown in Fig. 138, *23* the length of the splines is reduced and the groove is made of greater width b'. The lowering of the adjacent surface (Fig. 138, *24*) enables one to broach the splines more effectively and accurately.

Figure 138, *25*, *28*, *31* shows unsuitable shapes of tapering surfaces which do not allow overtravel and infeed of the tool. Correct designs are illustrated in Fig. 138, *26*, *27*, *29*, *30*, *32*, *33*. Figure 138, *34*, *35* shows irrational and Fig. 138, *36*, rational designs of spherical surfaces.

Let us discuss examples of wrong and correct designs of standard units and parts used in mechanical engineering.

In the design of a splined shaft with straight-sided splines (Figure 139, *1*) it is impossible to grind the working faces and the centring surfaces of the shaft. To permit overtravel of the grinding wheel the

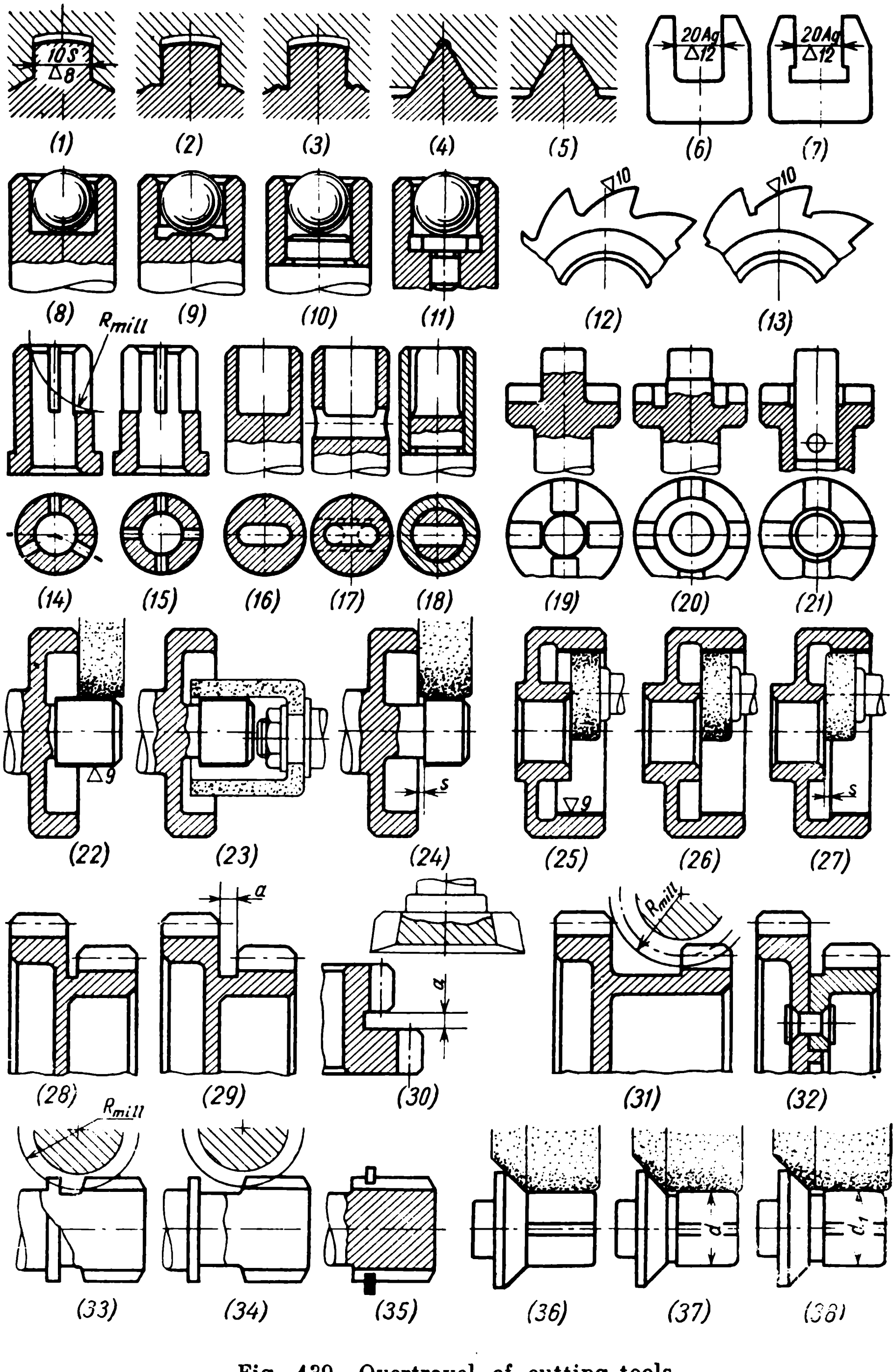


Fig. 139. Overtravel of cutting tools

surface of the shaft should be lowered at the base of the splines (Figure 139, *2*), or grooves should be made (Fig. 139, *3*).

Figure 139, *4*, *5* shows wrong and correct designs of an inverted V-guideway, respectively, and Fig. 139, *6*, *7*, those of a snap limit gauge.

The internal space of a step ball bearing (Fig. 139, *8*) can be machined easier if a groove is made at the base of the space (Fig. 139, *9*) or if use is made of composite structures (Fig. 139, *10*, *11*).

In the free wheel (Fig. 139, *12*) the spiral active surfaces of teeth (usually worked on relieving grinding machines) should be provided with undercuts to allow for overtravel of the grinding wheel (Figure 139, *13*).

It is impossible to mill the slots in the slotted bushing (Fig. 139, *14*) because the cutter comes against the bushing wall. If four instead of three slots are used (Fig. 139, *15*) they can be through-pass milled.

It is very difficult to machine the end slot in the shaft (Fig. 139, *16*). If the cutting tool overtravel is permitted into a transverse bore at the base of the slot (Fig. 139, *17*), the shaft end then can be drilled at the slot edges (dashed lines) and the partition between the drilled holes removed by planing. A composite design comprising a rim press-fitted onto the slotted portion of the shaft requires still simpler machining (Fig. 139, *18*).

End slots on a shaft (Fig. 139, *19*) can only be formed by upsetting. Separating the slots from the cylindrical surface of the shaft by an annular groove (Fig. 139, *20*) enables one to make them by planing. In the composite design (Fig. 139, *21*) the slots can be machined more accurately and efficiently by through-pass milling.

In the cup-shaped part (Fig. 139, *22*) the neck of the shaft can be ground only by a very expensive and inefficient method using a cup wheel mounted eccentrically with respect to the shaft (Fig. 139, *23*). To make cylindrical grinding possible the shaft journal should protrude beyond the cup to a distance s sufficient for overtravel of the wheel (Fig. 139, *24*).

In another cup-shaped part (Fig. 139, *25*), the grinding of the internal surface is hindered by the projecting end of the hub. The design in Fig. 139, *26* is also wrong because the end of the surface being ground coincides with the end of the hub, and a burr appears on the extreme portions of the surface.

In the correct design shown in Fig. 139, *27* the end of the hub is displaced relative to the surface being ground to a distance s thus ensuring a good finish of the entire surface.

In the cluster gear (Fig. 139, *28*) the teeth of the pinion can be cut if the distance a (Fig. 139, *29*) is made sufficient for overtravel of the gear cutter (Fig. 139, *30*). The minimum value of a (mm) as against the tooth module m is given below.

m	1-2	3-4	5-7	8-10	12-14
a	4-5	6-7	8-9	10	14

When teeth are formed by a hob cutter much larger distances are required, determined by the diameter of the cutter (Fig. 139, *31*) and the plan approach angle with respect to the shaft axis. If the rims have to be close together, composite designs are used (Figure 139, *32*).

To prevent the hob cutter from cutting into the thrust shoulder of the shaft (Fig. 139, *33*) when the splines are machined by the generating method, the shoulder must be positioned at such a distance from the shaft end as will permit the machining of the splines without the tool cutting into the shoulder (Fig. 139, *34*). The best way is to through-pass machine the splines and replace the shoulder with a circular stop (Fig. 139, *35*).

Figure 139, *36* shows a conical valve with a guiding shank. The valve chamfer and the centring surfaces of the shank are plunge-cut ground with a form wheel.

In this design it is impossible to finish grind the portion where the chamfer adjoins the shank. The design with a recess (Fig. 139, *37*) is also wrong because the diameter d of the shank is equal to the smaller diameter of the chamfer and a burr may appear on the chamfer.

In the correct design shown in Fig. 139, *38* the diameter d_1 of the shank is smaller than the minor diameter of the chamfer, and the surfaces of the shank and the chamfer being ground are overlapped by the grinding wheel.

4.7. Approach of Cutting Tools

To increase the efficiency and accuracy of the machining process the cutting tool should have an easy approach to the work surfaces. For this reason one must have a clear understanding of the machining operations, know the dimensions of the cutting tool and its fastening elements and the methods of mounting and clamping the work.

Figure 140, *1* presents a sheave of a V-belt transmission with a threaded hole n in the hub for the fastening screw. The shape of the part allows the hole to be drilled and threaded only through the bore m in the rim (Fig. 140, *2*) which should be provided in the design.

Some methods of making the hole n in a bracket (Fig. 140, *3*) are shown in Fig. 140, *4-6*.

When determining the inclination angle of a skew hole (Fig. 140, *5*), the drill chuck dimensions should be considered.

In the design of a pin-type fastening a cup-shaped part on a shaft (Fig. 140, *7*) it is impossible to drill and ream hole m for the pin and also insert the latter. In this case it is necessary either to provide hole m in the sheave rim (Fig. 140, *8*) or to change the position of the hub (Fig. 140, *9*).

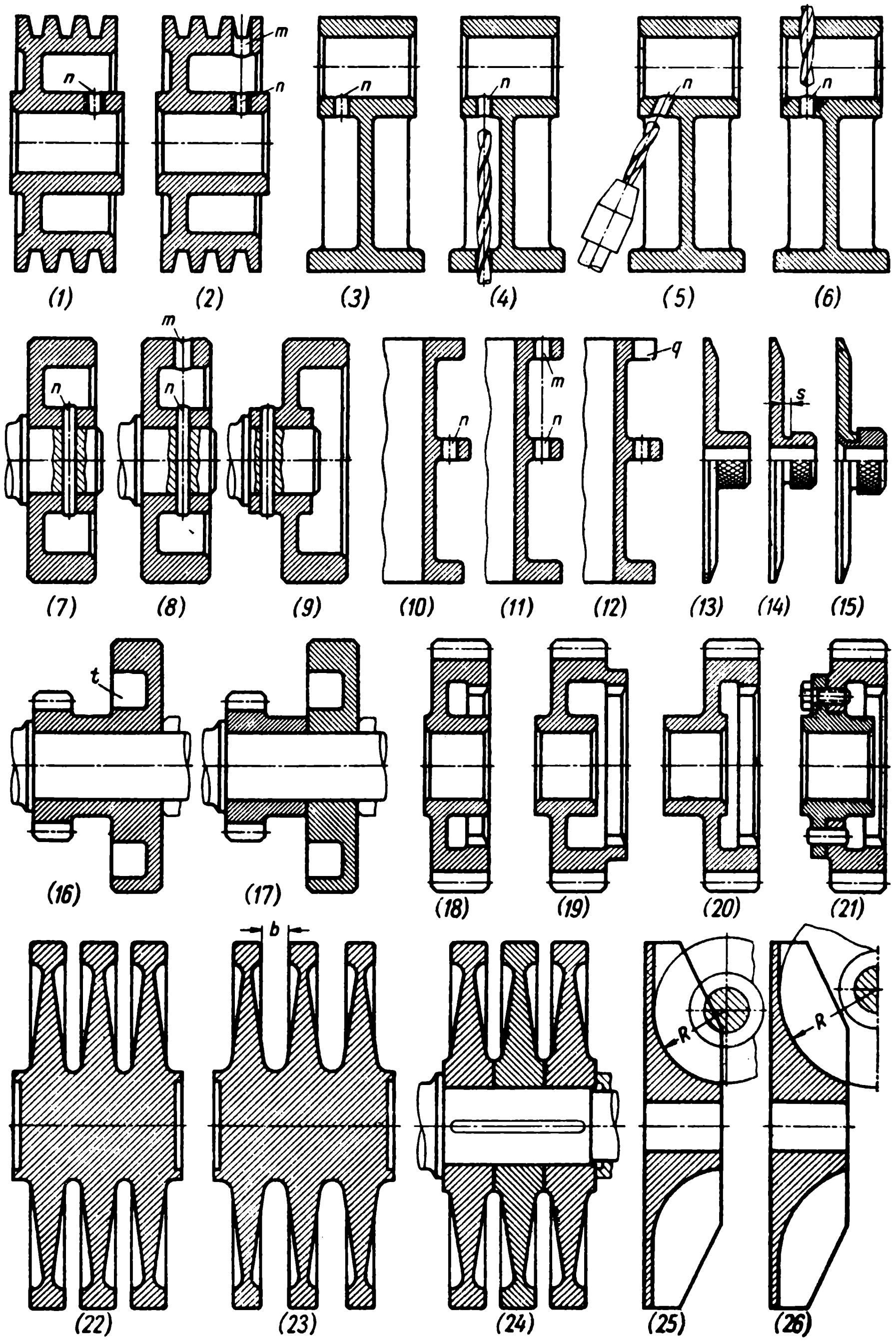


Fig. 140. Approach of cutting tools

Hole *n* (Fig. 140, *10*) in the l g between the flanges of a cylinder can be drilled through hole *m* (Fig. 140, *11*) or recess *q* in one of the flanges (Fig. 140, *12*).

When knurling the knob of the dial in the design shown in Figure 140, *13*, the knurling roller cannot reach the base of the knob. The knob should be displaced from the dial to a distance s = 3-4 mm (Fig. 140, *14*) sufficient to let pass the cheek of the roller holder.

When the dial is large in diameter a composite design (Fig. 140, *15*) is preferable, allowing the use of a short and rigid roller holder.

Shaped slot *t* in the face cam (Fig. 140, *16*) cannot be formed as it is impossible for an end mill to approach the slot because there is a gear made integral with the cam.

To make the machining possible, the cam must be made detachable from the gear (Fig. 140, *17*).

In the design of a gear with an internal splined rim (Fig. 140, *18*) the splines can be cut only by slotting. The more efficient and accurate generating method can be employed, if the splined rim is brought out beyond the hub Fig. 140, *19*), or if the hub is displaced (Fig. 140, *20*), or else if a composite design is employed (Fig. 140, *21*).

The internal faces of the disks in the one-piece turbine rotor (Fig. 140, *22*) can be machined if the disks are arranged farther apart by increasing distances *b* and reducing the width of the disk rims (Fig. 140, *23*), or if a split design (Fig. 140, *24*) is employed.

It is possible to mill the impeller blades of a centrifugal machine (Fig. 140, *25*) if the radius at the base of the blades is increased to an amount that permits approach of a milling cutter (Fig. 140, *26*).

Figure 141 shows examples of changes in design making the machining of hard-to-reach surfaces easier. The machining of inner space *m* of a stop valve housing (Fig. 141, *1*) can be simplified by increasing the diameter of the threaded portion of the housing (Figure 141, *2*). In this case, ordinary or core drilling may be used instead of turning on a lathe.

Figure 141, *3-5* shows the methods applied to facilitate the machining of internal space *n* of a turnable pipe connection.

The threads in holes should not be too deep (Fig. 141, *6*), but made as close as possible to the upper end face of the part (Fig. 141, *7*).

It is simpler to machine a labyrinth seal (Fig. 141, *8*) if the ridges are made outside of the seal housing (Fig. 141, *9*).

It is practically impossible to cut the thread on the rod of a cup-shaped part (Fig. 141, *10*). The machining can be done if the thread is cut beyond the cup (Fig. 141, *11*) or if a composite design (Figure 141, *12*) is employed.

The grinding of a deep hole in a shaft is illustrated in Fig. 141, *13*. The deflection and runout of the cantilever spindle carrying the grinding wheel make it impossible to obtain a well finished and

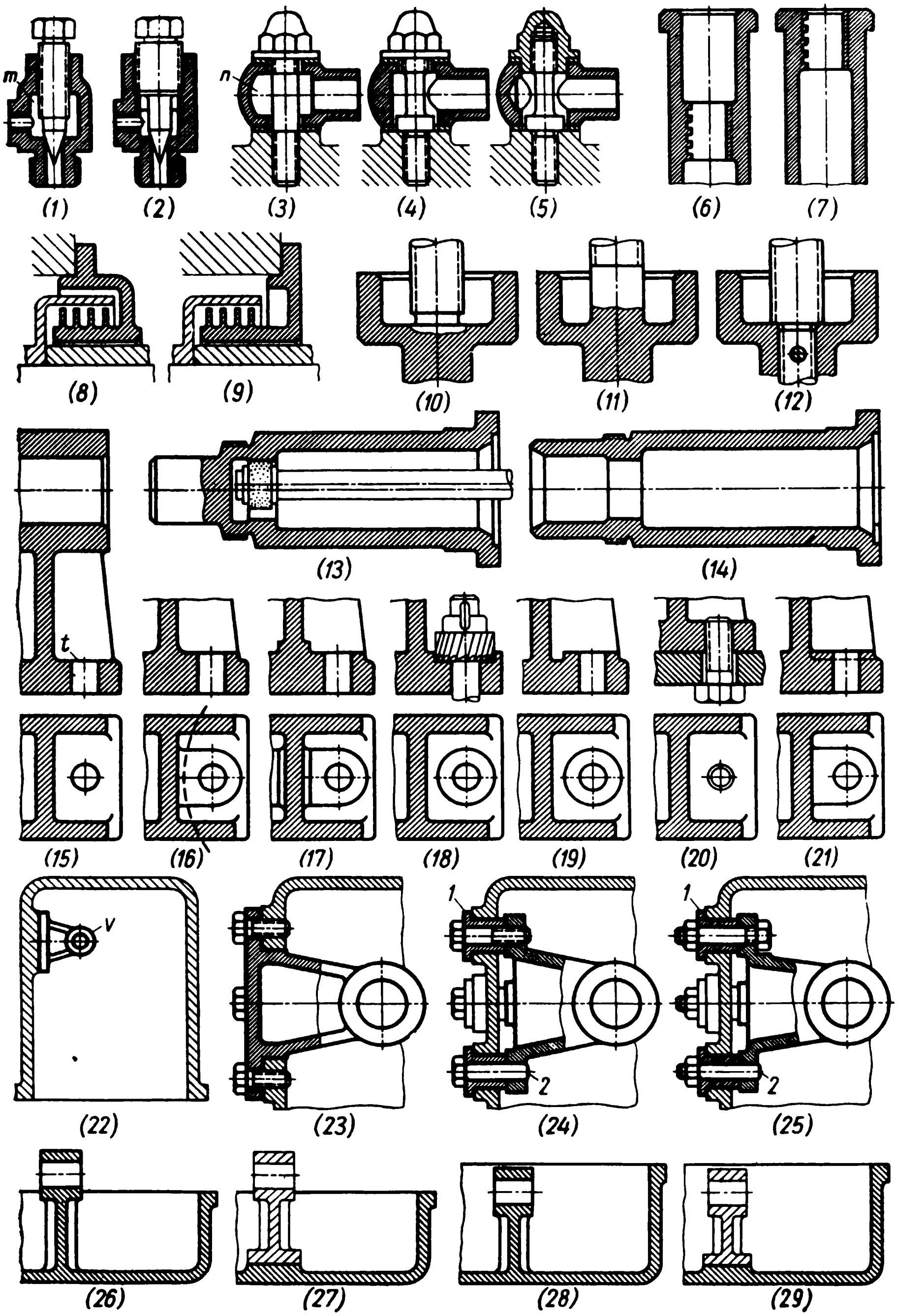


Fig. 141. Methods of making the machining easier

accurate surface. In the correct design shown in Fig. 141, *14* there is a through hole and the spindle now can be mounted on two supports (the shaft rotates in a chuck arranged eccentrically with respect to the spindle). With this design the grinding may be replaced by fine boring, reaming or broaching.

Figure 141, *15* shows difficult-to-machine surfaces *t* for fastening bolts in a bracket with a base connected by an H-section rib with a bushing.

Milling (Fig. 141, *16*) is impossible in this case because the ribs hamper approach of the milling cutter (dashed line). Planing (Fig. 141, *17*) is difficult since overtravel is not provided for the tool. Inverse spot facing (Fig. 141, *18*) can be applied only if the hole diameters are large.

The boss raised above the surface of the base can be planed (Figure 141, *19*) or the base can be secured with bolts (Fig. 141, *20*) mounted on the other side of the housing (in this case it is not necessary to machine the upper side of the base).

In the case of high-precision casting (for example, casting into metal moulds) the surface for nuts may be left rough (Fig. 141, *21*). However, the bearing surfaces in critical joints should be machined to prevent the skewing of the bolts.

It is extremely difficult to machine surfaces in deep cavities (pad for mounting part *v*, Fig. 141, *22*). The internal surfaces may be left unmachined, if the part is mounted on external pads and passed through a hole in the wall (Fig. 141, *23*).

If it is impossible to make the hole of the required size, the part is introduced into the cavity and fastened on bushings *1* (Fig. 141, *24*, *25*) flange-mounted on the outer pads of the housing, and the part being located in the bushings from set pins *2*.

Transverse holes arranged in housings at a considerable distance from the edges (Fig. 141, *26*) or in recesses (Fig. 141, *28*) can be machined only with an extended tool, a ratchet drill or, an angular drilling head, etc. In such cases it is more practical to use detachable brackets mounted on pads in the housing (Fig. 141, *27*, *29*).

4.8. Separation of Surfaces to Be Machined to Different Accuracies and Finishes

Surfaces to be machined with different tools and to different accuracies and finishes should be designed with some separating elements between them.

In a forked lug (Fig. 142, *1*) the surfaces of the slot and the base coincide. In the correct design (Fig. 142, *2*) the bottom of the slot is raised above the base surface to a distance *s* (at least by several tenths of a millimetre).

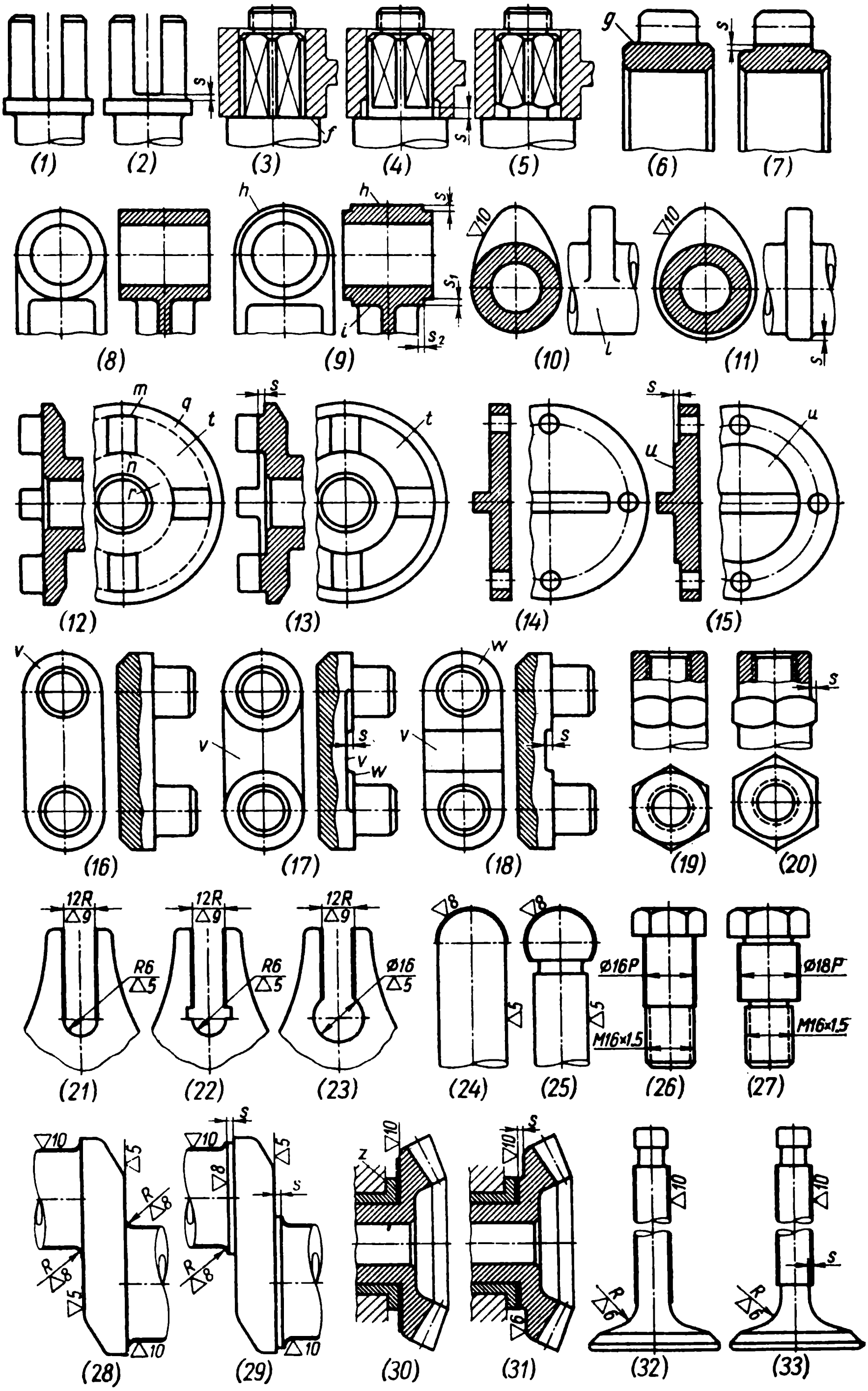


Fig. 142. Separation of surfaces to be machined by various methods

The design of a shaft with a square shank for a fitted-on part (Fig. 142, *3*) is wrong: it is practically impossible to machine the end face *f* of the shaft steplessly when the faces of the square are milled.

In the design shown in Fig. 142, *4* the faces are raised above the end face to a distance *s*. The face is undercut when the cylindrical surface of the shank is turned. On the fitted-on part a recess is provided to overlap the cylindrical shoulder.

The square of the shank can be separated from the shaft end face by an annular recess with a diameter slightly smaller than the distance between the square faces (Fig. 142, *5*).

In the wrong gear design (Fig. 142, *6*) the root surface of teeth coincides with cylindrical surface *g* of the gear rim. In the correct design shown in Fig. 142, *7* the root surface is raised above the hub surface to a distance *s* that ensures overtravel of the gear-cutting tool and prevents it from cutting into the rim surface.

It is practically impossible to manufacture a connecting rod end (Fig. 142, *8*) whose merging surfaces are machined by different operations.

In the design shown in Fig. 142, *9* the surfaces machined with different tools are separated. The external surface *h* of the H-section rod, which is machined with a plain milling cutter, is raised to a distance *s* relative to the connecting rod end. The internal spaces *i* of the rod, machined with a face cutter, are removed from the rod end to a distance s_1. The rod-end cantilevers, worked by turning, are separated from the rod by a distance s_2.

In the cam design (Fig. 142, *10*) the accurate surface of the cam merges with the cylindrical surface of the shaft which is machined to a lower accuracy. It is impossible to grind the back surface *l* of the cam flush with the shaft cylinder. In the correct design shown in Fig. 142, *11* the surface of the cam is raised above that of the shaft to a distance *s* ensuring the required machining of the cam.

In the dog plate design (Fig. 142, *12*) surfaces *m* and *n* of the dogs are turned together with annular sections *q* and *r* of the disk end face, and portions *t* are milled. It is impossible to match these surfaces. In the correct design shown in Fig. 142, *13* the surface to be milled is raised above the adjacent surfaces of the disk end face to a distance *s*.

Similarly, in the ridged plate design (Fig. 142, *14*, *15*) surface *u* to be milled should be higher than all the other surfaces of the end face which are turned.

It is difficult to machine the block with cylindrical pins (Fig. 142, *16*). It is necessary to turn surfaces *v* adjoining the pins in two cuts so that the surfaces are matched precisely. The design with cylindrical bases *w* raised to a distance *s* (Fig. 142, *17*) is correct only if the surface *v* of the block between the pins can be left rough, because it is difficult to machine this surface.

If the surface adjoining the pins is to be machined, it should be shaped as shown in Fig. 142, *18*. The bases *w* of the pins are turned on a lathe and the surface *v* is through-pass milled.

In hexagons adjoining cylindrical surfaces (Fig. 142, *19*) the faces should be arranged above the cylindrical surface (Fig. 142, *20*).

In the design shown in Fig. 142, *21* it is impossible to merge the ground working faces of the slot with its drilled base. The precision- and rough-machined surfaces should be separated (Fig. 142, *22*) or the base of the slot drilled to a diameter larger than the slot width (Fig. 142, *23*) to ensure overtravel of the grinding wheel.

Examples of wrong and correct merging of accurate and rough surfaces are illustrated in Fig. 142, *24*, *25* (push rod with a spherical head) and *26*, *27* (dowel bolt).

The design of the joint between the crankpin, main journal and webs of a crankshaft (Fig. 142, *28*) is erroneous: the ground fillets of the journals pass directly into the milled webs. In the correct design shown in Fig. 142, *29* the fillets are separated from the web surfaces by shoulders *s*.

In the bevel gear (Fig. 142, *30*) the ground bearing surface *z* passes into the turned fillet of the end surface of the teeth. It is practically impossible to obtain the smooth mating shown on the drawing. In the correct design (Fig. 142, *31*) the surface to be ground is separated from the rough surface by step *s*.

In the disk valve (Fig. 142, *32*) the guiding surface of the rod, machined to a high accuracy and finish, gradually forms the fillet of the head. This fillet can be obtained in practice only by filing manually the transition section. In the correct design shown in Fig. 142, *33* the surface of the rod is separated from the fillet by a recessed portion *s*.

It is expedient to separate cylindrical surfaces of the same diameter machined to different classes of finish (Fig. 143*a*) by a shallow groove (Fig. 143*b*) or to through-pass machine the entire surface to the same finish.

Surfaces having the same nominal diameter, but machined to different tolerances so as to ensure different fits (Fig. 143*c*) should preferably have their seating sections separated by a groove (Fig. 143*d*), or one of the sections should be made of a smaller diameter than the other (Fig. 143*e*).

If the nominal diameter of the seating surface on a shaft is equal to the major diameter of the adjacent thread (Fig. 143*f*), an increase in the thread diameter (due to the threads' "rising" during cutting) often makes it impossible to install the fitted-on part on the shaft.

In such cases the major diameter of thread should be through-pass machined together with the seating surface, a special note being made for the purpose on the drawing. But it is better to reduce the thread diameter (Fig. 143*g*).

Figure 143*h* shows wrong and Fig. 143*i*, *j*, correct designs of separating internal cylindrical surfaces machined to different classes of finish.

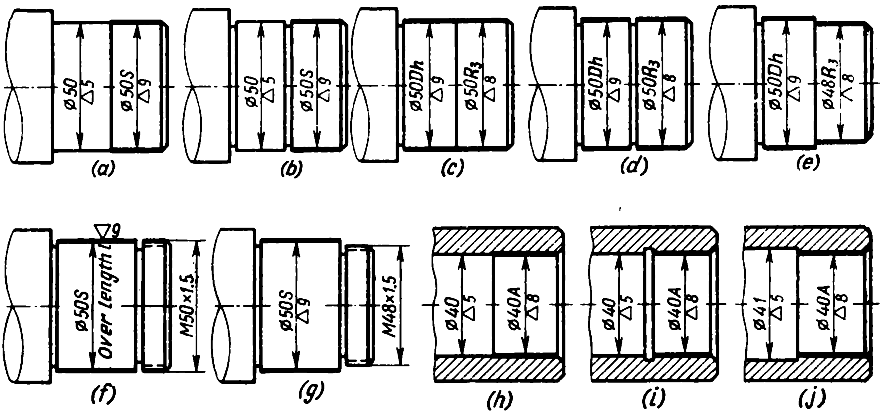


Fig. 143. Separation of surfaces machined to different finish for various fits

4.9. Making the Shape of Parts Conformable to Machining Conditions

The shape of parts to be machined must conform to the type of machining, the shape and size of the cutting tool, and the sequence of operations.

Figure 144 shows a connecting rod end joined to an H-section rod. The design shown in Fig. 144*a* can be obtained only by closed-impression die forging and cannot be machine cut. With the shape shown

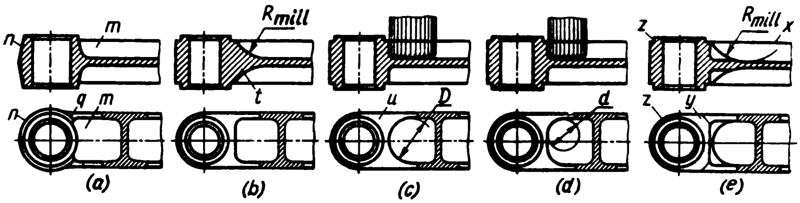


Fig. 144. Joining a connecting rod end to an H-section rod

on the drawing, the recess *m* between the flanges cannot be milled. The contour machining of the external surface *n* of the end and the sections *q* where the flanges pass into the end is likewise impossible.

The recess can be milled with a plain cutter (Fig. 144*b*) or with a face cutter (Fig. 144*c*). Both methods fully determine the shape of the joint, which must be shown on the drawing.

Heavy sections *t* (Fig. 144*b*) and *u* (Fig. 144*c*) at the joint between the rod and its end are eliminated by face milling the transition portions (Fig. 144*d*, *e*).

Ends *x* of the flanges are milled with a face or plain milling cutter up to surface *y* which is undercut when turning ends *z* of the bushings.

The conjugation of a round bar and a forked lug (Fig. 145*a*) cannot be machined and is only obtainable by closed-impression die forging.

In the design in Fig. 145*b*, the bar is turned, and the lug, milled. In the esign in Fig. 145*c*, the lug takes a cylindrical shape, and only

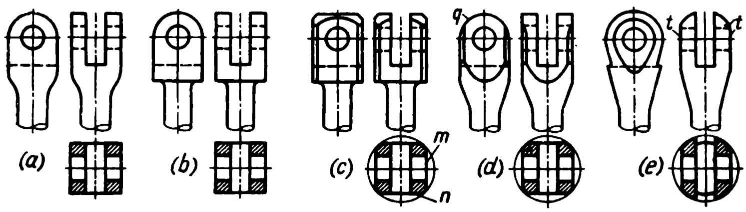


Fig. 145. Machining of a forked lug

faces *m* and *n* are milled. In the design with the lug tapering towards the bar (Fig. 145*d*) the taper and cylinder surfaces are turned, and the side faces and rounded end *q*, milled.

In the most rational design (Fig. 145*e*) the lug having the shape of a sphere with a taper towards the bar is turned on a lathe and only side faces *t* are milled.

4.10. Separation of Rough Surfaces from Surfaces to Be Machined

On blanks produced by casting, stamping, forging, etc., the work surfaces must be separated from the nearest rough surfaces by a distance *k* exceeding the amount of possible displacement of the rough surfaces.

Figure 146 illustrates the application of this rule to work surfaces arranged above (Fig. 146*a*) and below (Fig. 146*b*) rough surfaces, and also to those adjacent to rough walls (Fig. 146*c*).

If the distance *k* is insufficient, an upward displacement of the rough surface in casting (Fig. 146*a*) will cause the tool to cut into the wall, and in the case of a downward displacement the tool will fail to reach the wall leaving it rough. In Fig. 146*b*, if the rough surface is displaced downwards, the tool may not reach the metal. The displacement of side walls (Fig. 146*c*) may cause the tool to cut into the wall metal.

Figure 146*d-f* shows this rule as applied to separating the work surfaces on fastening flanges.

Sometimes, dimensions do not allow rough walls to be removed from the work surfaces. In such cases the required distance *k* can be

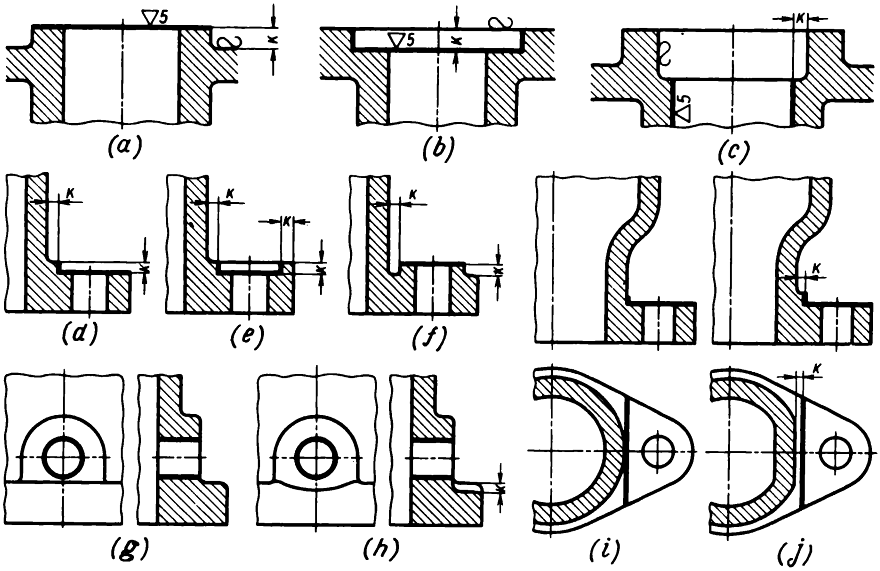


Fig. 146. Separating of rough surfaces from surfaces to be machined

maintained by making local recesses, cavities, etc. in the walls (Fig. 146*g*, *i*—wrong designs, Fig. 146*h*, *j*—correct designs).

The value of *k* mainly depends on the manufacturing accuracy of the blank and its overall dimensions. The values of *k* for cast parts can be found from Fig. 122.

For parts made by smith forging the values of *k* are about the same. In the case of die-forged parts, *k* varies within 0.5 to 2-3 mm, depending on the forging accuracy and dimensions of the blank.

Figure 147*a* shows a case of facing a boss on an internal wall of a cast housing, effected through a hole in an external wall. The diameter of the hole in the external wall is equal to the diameter *d* of the boss. If the boss is displaced from its nominal position in casting, an unmachined burr may appear on the boss. In this design the end face can be machined only with the aid of a boring bar with an extensible tool.

The correct design is illustrated in Fig. 147*b*. The diameter of the hole in the external wall is made larger than the boss diameter by the amount 2*k* of possible displacements.

In the design shown in Fig. 147*c* the faced surface of the boss is arranged below the rough surface, and the diameter of the boss is increased. As a result, the facing tool cuts a correct cylindrical surface in the boss.

Figure 147*d* shows the spot facing of a boss in a pit with rough walls. The size of the pit does not permit the use of a spot facer of such a

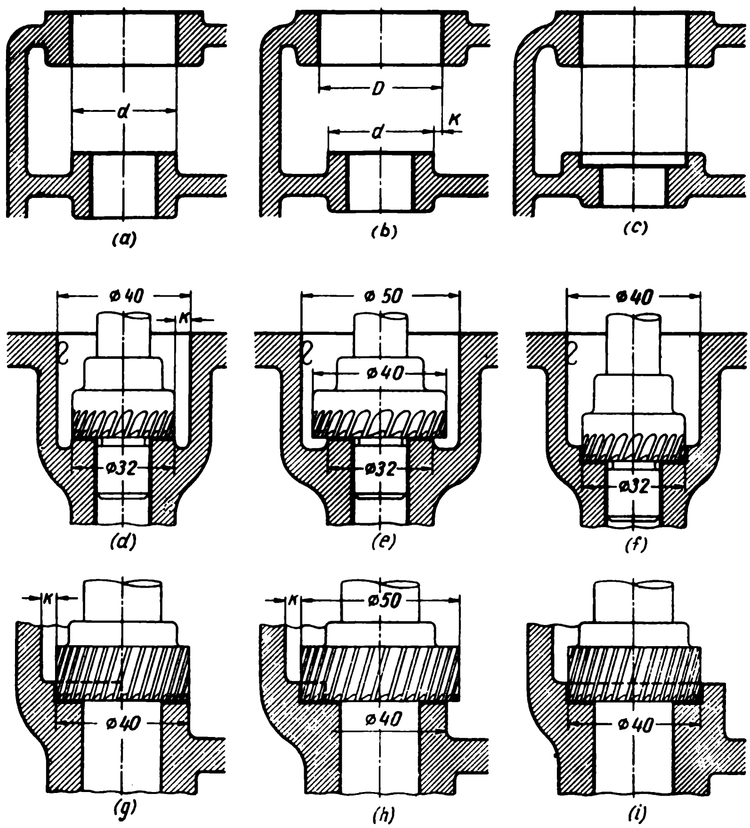


Fig. 147. Facing of bosses

diameter as is required to correctly machine the boss and keep at the same time proper clearance k between the spot facer and the walls of the pit.

In the design shown in Fig. 147*e* the diameter of the pit is increased so that the boss is overlapped by the spot facer. In the design in Fig. 147*f* the work surface is sunk in the bottom of the pit.

Figure 147*g*-*i* illustrates the facing of a boss adjoining the wall of a part (Fig. 147*g*—wrong design, Fig. 147*h*, *i*—correct designs).

4.11. Machining in a Single Setting

Surfaces which require precise mutual coordination should be machined in one setting.

In the speed reducer with overhung gears (Fig. 148*a*) the holes for the input and output shafts are machined from different sides of the

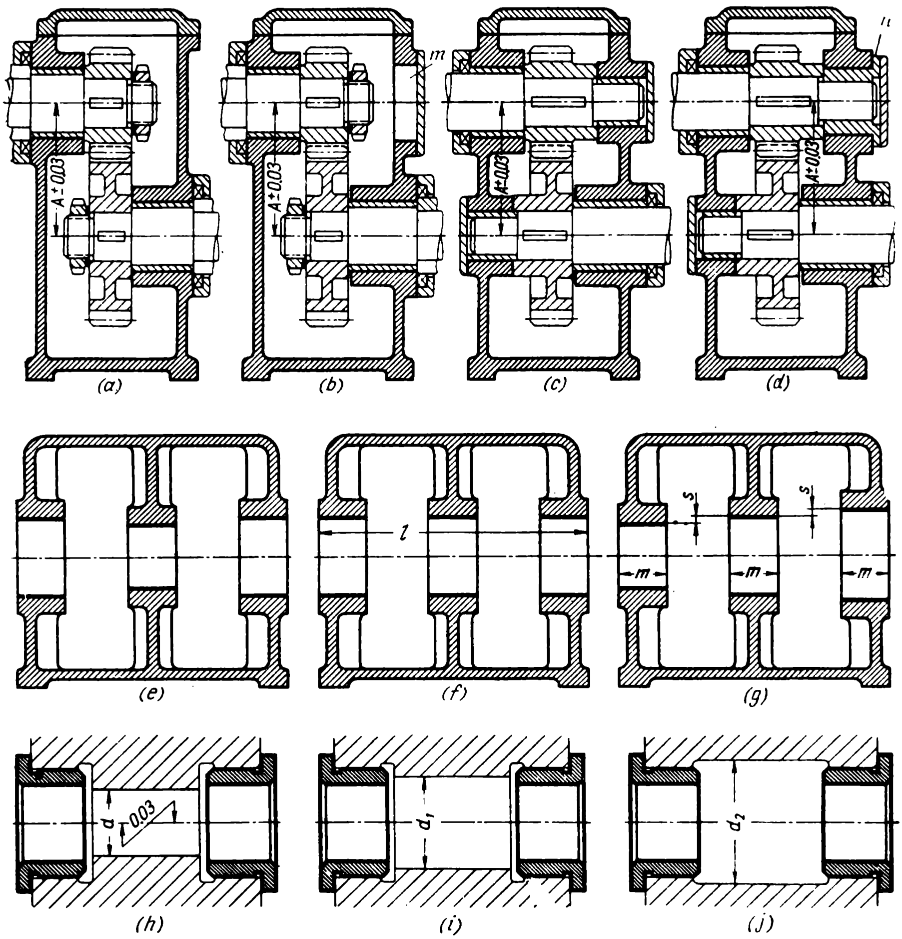


Fig. 148. Machining in a single setting

housing. In this case it is difficult to maintain centre distance *A* and make the hole axes strictly parallel.

In the good design shown in Fig. 148*b* provision is made for an additional hole *m* which makes it possible to machine the seating holes from one side.

In the speed reducer with stepped holes for the doubly-supported gears (Fig. 148*c*) the hole steps are wrongly arranged and cannot be machined from one side.

In the correct design shown in Fig. 148*d* an idle bushing *n* makes it possible to machine the holes from one side.

It is difficult to align to holes in the housing (Fig. 148*e*) because the small diameter of the middle hole hampers the through-pass machining of the holes.

Holes of the same diameter (Fig. 148*f*) or stepped holes of a diameter diminishing in the direction of the cutting tool run (Fig. 148*g*) are preferable for housings. The latter design is simpler and the efficiency of machining in this case is higher. If the difference *s* between the radii of the adjacent holes is larger than the machining allowance, the stroke of the boring bar with respect to the work is reduced to a magnitude slightly greater than the maximum width *m* of the holes being machined, and all the holes are machined simultaneously.

In the design with holes of the same diameter (Fig. 148*f*) the boring bar stroke is many times longer and must exceed the distance *l* between the extreme points of the surfaces being machined.

Holes of the same diameter can effectively be machined by means of boring bars with extensible tools which are set to the required size after introducing the boring bar into the blank.

In the unit with bushes mounted in a housing (Fig. 148*h*) the seating surfaces for the bushes can be machined only from the different sides of the housing because the diameter *d* of the intermediate hole is small. It is difficult to obtain proper axial alignment of the holes.

In the improved design shown in Fig. 148*i* the diameter d_1 of the intermediate hole is increased to the size which allows the press-fitted bushes to be reamed simultaneously.

The design in Fig. 148*j* is most advisable. Here, the diameter d_2 of the intermediate hole is increased to such a size as makes it possible to through-pass machine the seating holes for the bushes and ream them together.

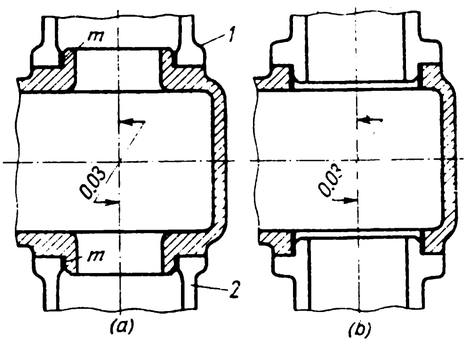


Fig. 149. Centring of parts in a housing

Figure 149 shows the centring of parts *1* and *2* arranged on the different sides of a housing. In the design shown in Fig. 149*a* the centring surfaces *m* are made in the form of collars on the housing, and it is practically impossible to align them.

In the design in Fig. 149*b* the centring is effected from holes in the housing which are machined in a single setting, this ensuring complete alignment of the parts being centred.

When machining the housing for rolling-contact bearings (Fig. 150) it is necessary to keep the alignment of the centring surface *m* of the

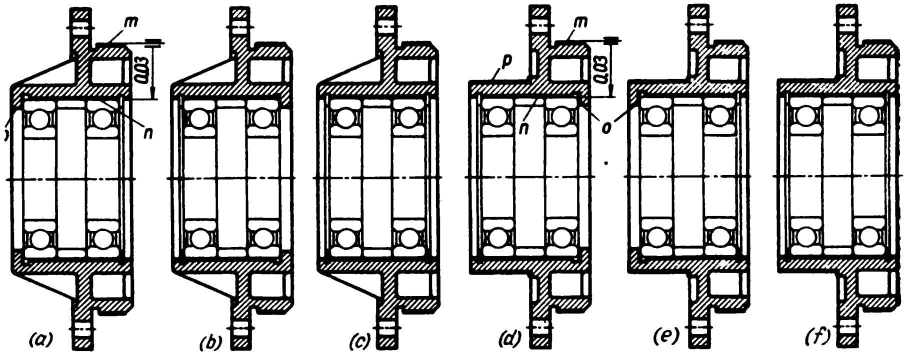


Fig. 150. Machining of concentric surfaces

housing and the seating surfaces *n* for the bearings to the given strict tolerances.

This can be attained by either of the following two methods:

(1) the housing is located on a mandrel from surface *n* finish machined in advance and then surface *m* is machined;

(2) the housing is clamped in a chuck on finish machined surface *m* and then surface *n* is machined.

Neither method can be applied with the design shown in Fig. 150*a* because the thrust shoulder *o* is arranged wrongly. Such a possibility occurs if the shoulder is transferred to the right-hand side of the housing (Fig. 150*b*) or replaced by a stop ring (Fig. 150*c*).

Surfaces *m* and *n* can be made concentric more simply and accurately, if the part is clamped in a chuck on surface *p* machined previously and the surfaces then machined in a single setting. In this case it will be wrong to arrange the thrust shoulder *o* on the right (Figure 150*d*). For correct machining the shoulder should be transferred to the left (Fig. 150*e*) or replaced by a stop ring (Fig. 150*f*).

4.12. Joint Machining of Assembled Parts

The joint machining of assembled parts should be avoided, for this complicates and splits the flow of production and spoils the interchangeability of parts in a given design.

Exceptions to this rule are the cases when the joint machining is the only method that can ensure the operating ability of the design. Thus, for example, in the case of multiple-bearing crankshafts, the splitting of the crankcase along the bearing axis is a prerequisite for assembly, and the joint machining of the bearing seat halves in the assembled crankcase is the only method to ensure the alignment of the bearings. The housings of rotory-type machines are frequently made split along the axis to facilitate assembly and disassembly and simplify inspection.

The joint machining of the internal surfaces and end faces of the bearing seats is required in the gear drive housing split along the shaft axis (Fig. 151*a*). Prior to the machining of the bearing seats,

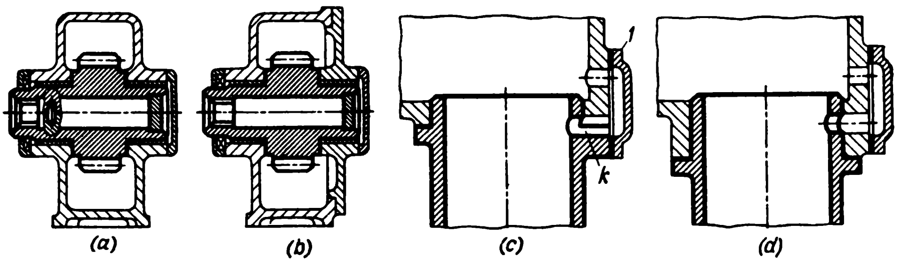


Fig. 151. Combined machining in assembly

the jointing faces of the housing halves must be finish machined and the halves positioned properly with respect to each other by means of set pins. The sealing of the joint with a gasket in this case is impermissible, and the butt-jointed surfaces are ordinarily lapped-in, the design losing its property of interchangeability of parts. Only the jointly machined housing halves can be accepted for assembly. It is impossible to replace a housing half during operation as this disturbs the cylindricity of the bearing seats and the alignment of their end faces.

The parts of the housing split in a plane perpendicular to the shaft axis (Fig. 151*b*) can be machined separately. The manufacture of the housing is greatly simplified, and the housing parts are interchangeable.

Figure 151*c* shows the cylinder of a rotary filler mounted on a tank. The cavities of the cylinder and tank communicate through by-pass hole *k*. Two errors are committed in this design: (1) the hole is drilled simultaneously in the cylinder flange and the tank body; and (2) cover *1* enclosing the by-pass holes is mounted at the joint between the cylinder flange and the tank wall. It is necessary to machine the hole and the joint surface together when the cylinder is assembled with the tank. The cylinder cannot be replaced during operation.

In the correct design shown in Fig. 151*d* the holes in the tank and the cylinder can be drilled separately. The joint surface is provided on the tank wall, and the cylinder can be replaced even when machined to ordinary accuracy.

4.13. Transferring Profile-Forming Elements to Male Parts

Internal surfaces are much more difficult to machine than external ones, and for this reason it is good practice to arrange profile-forming element on external surfaces. Figure 152*a*, *b* illustrates a labyrinth seal. The ridges made on the male part (Fig. 152*b*) are much simpler to manufacture than those in the hole (Fig. 152*a*).

The needle bearing in which the retaining shoulders are provided on the inner race (Fig. 152*d*) is better from the viewpoint of manufacture than the one with the shoulders on the outer race (Fig. 152*c*) since the hole in the outer race in this case is through-pass machined.

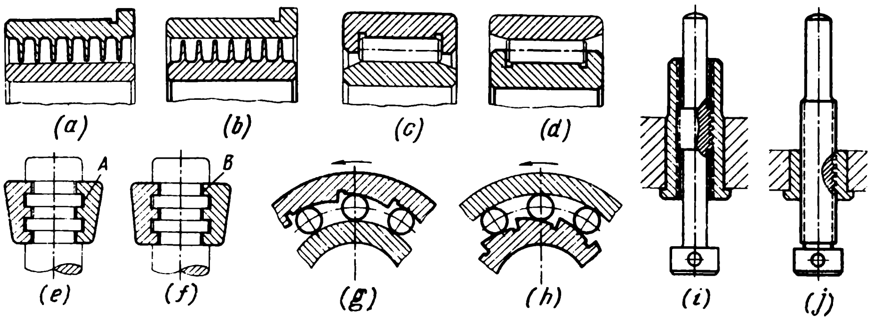


Fig. 152. Transferring profile-forming elements to male parts

The design of the unit for fastening a spring cap on a valve rod by means of split tapering blocks centred by the outer cylindrical surfaces *A* of the ridges (Fig. 152*e*) is irrational. The sound design is the one in which the accurate centring surfaces *B* are through-pass machined in blocks (Fig. 152*f*).

In a roller overrunning clutch the profiled elements (usually having the shape of a logarithmic spiral) should not be arranged on the outer race (Fig. 152*g*). They can be machined only by broaching and only when the hole in the race is a through one. In the design shown in Fig. 152*h* the external profiled elements can easily be processed, for example, on a relieving lathe.

Long threads in holes should be avoided (Fig. 152*i*). A long thread is good on a bar and a short one in a bushing (Fig. 152*j*).

4.14. Contour Milling

Complex and irregular profiles are more difficult to mill than flat or cylindrical surfaces.

The lever design requiring an all-round machining (Fig. 153*a*) is bad. The re-entrant angles do not permit the external contour of the

part to be machined with a plain milling cutter. It is also very difficult to machine surfaces *m* confined within the cylindrical walls of the bosses.

In the design shown in Fig. 153*b* the external contour is described by straight lines and circumferences and can be form milled. Sections *n* between the bosses, which are bordered by straight lines, can be through-pass milled. One side of the lever (surface *p*) is made flat to simplify machining.

It is practically impossible to mill the contour of the flange (Fig. 153*c*) because the fillets at the base of the bosses are too small.

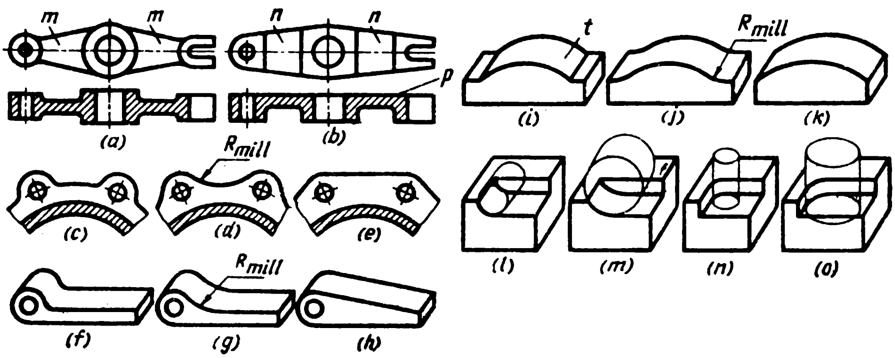


Fig. 153. Contour milling

The portions between the bosses should be profiled to a radius at least equal to the radius of the milling cutter (Fig. 153*d*) or along straight lines (Fig. 153*e*).

Figure 153*f* shows a wrong and Fig. 153*g*, *h*, correct designs of a lever requiring circular milling.

The design of the block in Fig. 153*i* is technologically incorrect: the cylindrical contour *t* can be machined only with a form milling cutter and cross blank feed, or by form planing.

In the more practical design shown in Fig. 153*j* the cylindrical surface is connected with the side flanges by a fillet having its radius equal to the radius of the milling cutter, which allows this surface to be milled with a standard plain cutter and longitudinal blank feed.

In the design shown in Fig. 153*k* the entire surface of the part is made cylindrical. The part can be milled in a swivelling fixture or turned in an attachment.

The milling efficiency and durability of milling cutters can be increased, if the cutters of the maximum permissible diameter are employed.

When machining the flat recess (Fig. 153*l*) the assigned contour of the recess can be obtained only with a small-diameter end milling cutter on a vertical-milling machine. The inadequate rigidity of the cutter makes it impossible to obtain a correct surface.

In the design shown in Fig. 153*m* the surface is machined with a larger cutter mounted on a doubly-supported spindle (milling machine).

Machining with an end mill (Fig. 153*n*) is allowed only as an exception, when a surface has to be imparted a nearly rectangular shape. This is a very ineffective method and it is impossible to obtain a well finished surface.

Figure 153*o* shows a case of machining with an end cutter of increased diameter, which overlaps the surface being milled.

4.15. Chamfering of Form Surfaces

The chamfering of form surfaces should be avoided. Form milling with a special cutter is required to chamfer the flange shown in

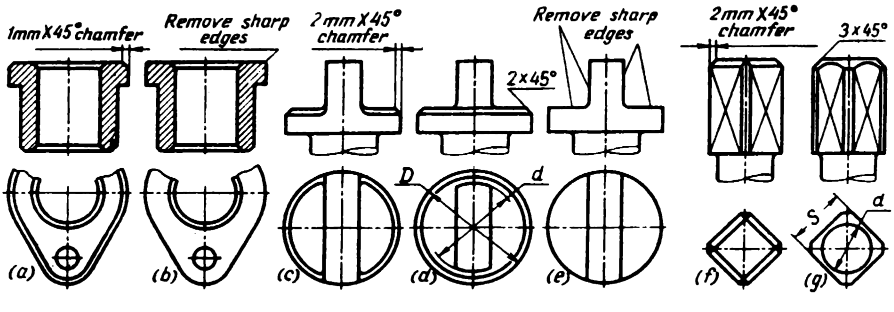


Fig. 154. Chamfering form surfaces

Fig. 154*a*. It is more advisable to break the corners (Fig. 154*b*), an operation carried out more simply (especially by the electrochemical etching method).

The chamfering of the face cam base (Fig. 154*c*) is simplified, if the diameter *d* of the cylindrical portion of the cam is reduced relative to the base diameter *D* by an amount exceeding the doubled chamfer width (Fig. 154*d*). If, for design considerations, diameter *d* cannot be decreased, it is then necessary to simply remove all sharp corners (Fig. 154*e*).

The chamfering of the square faces (Fig. 154*f*) requires a special milling operation with many resets of the part in the process of machining. In this case it is more practical to mill the faces on a previously turned cylinder with an end chamfer whose minor dia-

meter d must be smaller than the distance S between the faces (Figure 154*g*). The chamfers at the corners where the faces meet will then be the traces of the previous machining of the cylinder.

4.16. Machining of Sunk Surfaces

Contour milling with cutting into a rough surface (Fig. 155*a*) should be avoided. Such surfaces can be machined only with an end milling cutter whose diameter corresponds to the minimum radius R of the

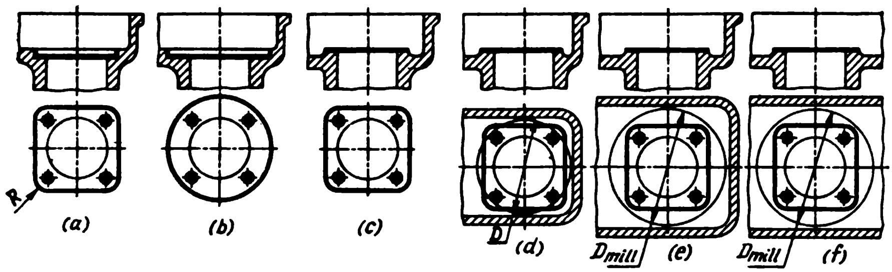


Fig. 155. Milling of sunk surfaces

rounded portion of the surface. The surface has to be machined in several cuts, the operation is ineffective, and it is impossible to obtain good surface finish.

The machining can be simplified if the surface is given a round shape with a diameter exceeding the maximum diagonal of the form surface (Fig. 155*b*). Such a surface can easily be face milled. A shaped flange can then be connected to it.

It is better to make the form surface as a pad protruding above the rough surface (Fig. 155*c*) and machine the pad with a face cutter.

The design should ensure the use of a milling cutter that overlaps the entire work surface.

In the design shown in Fig. 155*d* the latter condition is not satisfied: the maximum diameter D of the milling cutter, limited by the adjacent walls, is insufficient and the surface has to be machined in several passes with a cutter of a smaller diameter.

In the design in Fig. 155*e* the walls are brought farther apart by an amount that permits the entire surface to be overlapped by the cutter. Machining is done with infeed, moving the blank in the direction perpendicular to the work surface.

Through-pass machining with a longitudinal feed (Fig. 155*f*) gives the best results as to the machining efficiency and surface finish.

4.17. Machining of Bosses in Housings

The machining of internal end faces of holes in housings (Fig. 156*a*), counterboring (Fig. 156*b*) and chamfering (Fig. 156*c*) are rather difficult operations.

In housings with blind walls such surfaces can only be machined by means of boring bars with extensible tools. Boring bars of usual

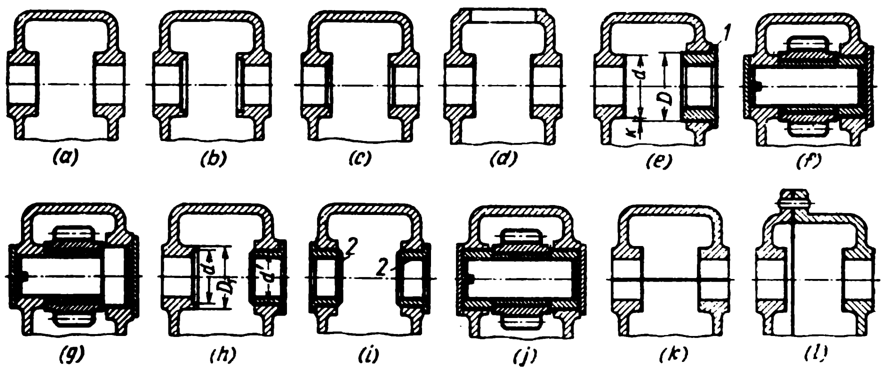


Fig. 156. Machining of bosses in housings

design can be used if an aperture is provided near the holes (Fig. 156*d*) for mounting the tools.

In order to increase the machining efficiency the diameter of the hole on the side where the tool is admitted (Fig. 156*e*) is made larger than the diameter of the boss of the second hole by the amount $2k$ of the maximum possible displacements of the boss in casting. In this case the end face of the smaller hole is machined with a spot facer. The other end bearing surface is obtained by inserting bushing *1* into the larger hole.

The design of a unit for such a case is presented in Fig. 156*f* (mounting of an idle gear wheel). Another design is also possible: a stepped shaft with the wheel resting against the shaft shoulder (Fig. 156*g*).

When counterboring the end face of the smaller hole (Fig. 156*h*) the diameter d' of the larger hole must not be smaller than the counterbore diameter d. To prevent the formation of weak thin edges the diameter D_1 of the rough surface of the boss should exceed the counterbore diameter d by not less than 8-10 mm.

Instead of facing, adapter sleeves *2* (Fig. 156*i*) may be employed the ends of which can serve as bearing surfaces (Fig. 156*j*).

In housings split along the axis of holes (Fig. 156*k*) the same rules should be observed because the end faces should be machined together after both halves of the housing are assembled.

In housings where the parting plane is perpendicular to the axis of holes (Fig. 156*l*) the holes are machined with the halves assembled and located one with respect to the other by set pins. The end faces of the bosses can be machined when the housing halves are detached.

4.18. Microgeometry of Frictional End Surfaces

The frictional end surfaces of holes should preferably be machined by the methods involving the rotation of the tool (or the part) about the hole centre (turning, boring, counterboring). The microscopic lines left after such machining are oriented more favourably with respect to the direction of the working motion than the longitudinal

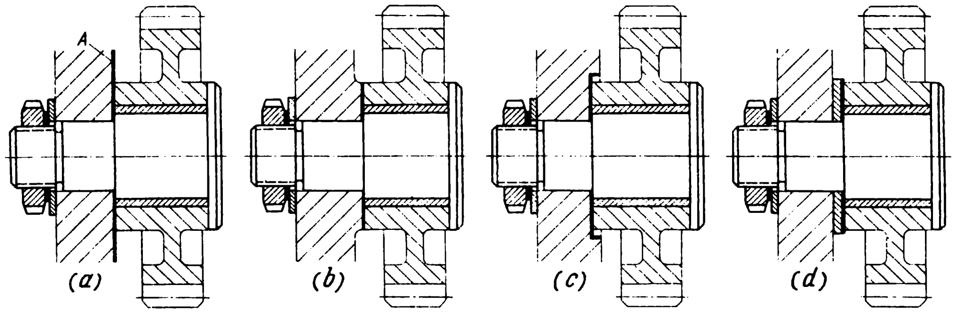


Fig. 157. Machining of frictional end surfaces

or transverse lines formed by planing and milling. Surfaces machined by this method are run in much faster. Besides, with such a machining it is easier to ensure the squareness of the frictional surface with the rotation axis.

The design of a gear wheel unit mounted on a housing wall where the wheel rests against the milled surface *A* (Fig. 157*a*) is irrational. It is better to spot-face (Fig. 157*b*) or counterbore (Fig. 157*c*) the frictional surface. It is also possible to mount a bearing ring washer (Fig. 157*d*).

4.19. Elimination of Unilateral Pressure on Cutting Tools

When machining holes with cylindrical tools (drills, counterbores, reamers) it is necessary to prevent unilateral pressure on the tool, which impairs machining accuracy, intensifies wear and sometimes causes breakage of the tool.

In the design shown in Fig. 158*a* the tool at the section *m* cuts into the rough vertical wall of the product. During the process of machining the tool is subjected to a unilateral pressure, and the hole deflects to the opposite side.

The design in Fig. 158*b* is better. The tool experiences a unilateral pressure only during the last machining stages.

Proper machining conditions will be ensured when the tool engages the metal with its whole surface. For this purpose the end of the hole should be positioned below the rough surface (Fig. 158*c*) or raised above it (Fig. 158*d*).

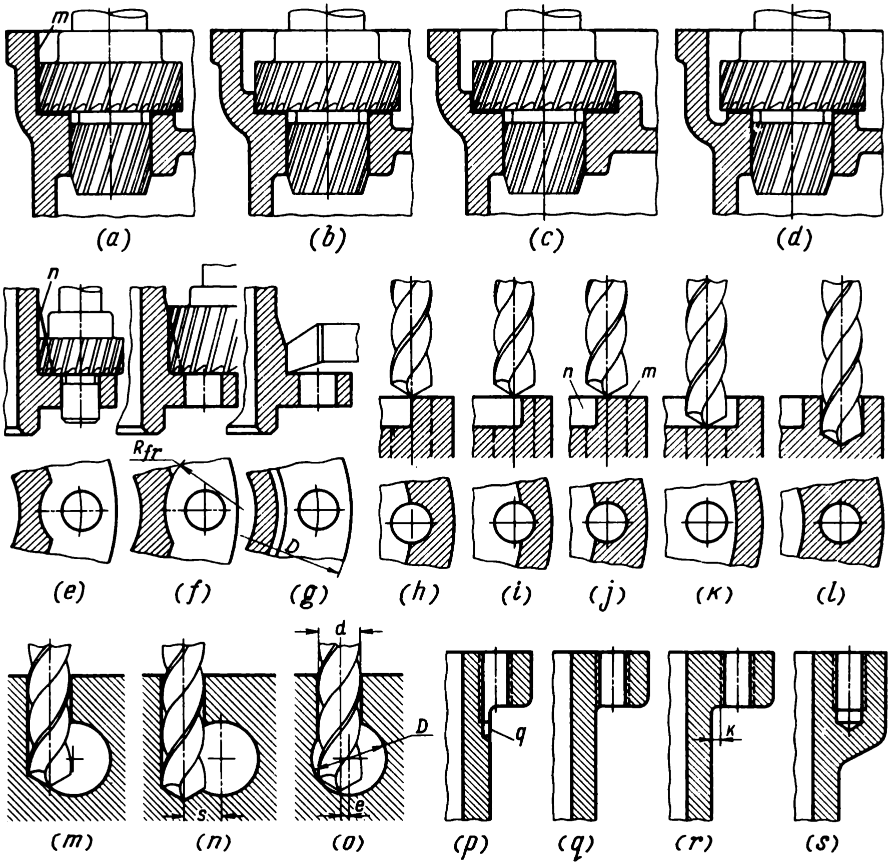


Fig. 158. Elimination of unilateral pressure on cutting tools

When spot-facing the fastening holes of a steel flange (Fig. 158*e*), cutting into the taper *n* which connects the flange with the cylinder walls will displace the tool mainly because the dimensions of the part do not allow the tool to be secured on a rigid arbour. If the shape of the flange is not changed, the flange has to be machined with a cutter of an increased diameter mounted on a rigid arbour advanced sideways (Fig. 158*f*). It is likewise possible to increase the diameter *D* and machine the flanges by turning (Fig. 158*g*).

Figure 158*h-l* illustrates the arrangement of holes on a stepped surface. The holes intersecting the step (Fig. 158*h-j*) can be drilled only with the aid of a jig. It is possible first to drill holes through the previously machined surface *m* (Fig. 158*j*) and then turn the recess *n*. But this method disturbs the sequence of turning operations.

It would be better to offset the holes to one or the other side of the step (Fig. 158*k*, *l*). In this case the drilling can be done without disturbing the sequence of turning operations. The offset should be large enough to prevent the formation of a thin partition between the drilled hole and recess (Fig. 158*l*).

Holes with intersecting axes should be avoided as far as possible. It is bad when the centre of the drill presses against the inclined wall of a transverse bore (Fig. 158*m*). It is somewhat better when the vertical bore is offset with respect to the axis of the cross drill by an amount s sufficient to centre the drill over the entire cutting path (Fig. 158*n*).

It is good practice to drill the hole through the centre of the transverse hole or with an offset e relative to it (Fig. 158*o*). The maximum value of e with which the drill functions properly can be found from the formula $e = 0.2\,D\left(1 - \frac{d}{D}\right)$.

If D considerably exceeds d the vertical hole can be drilled first, and then the transverse one. In this case the amount of offset e is immaterial.

It is also recommended to ensure cutting over the entire hole circumference at the exit of the tool.

In Fig. 158*p* the threaded hole in the flange in section *q* cuts into the wall of the part and the tool (drill and tap) is subjected to a unilateral pressure, which may cause its breakage.

In the design shown in Fig. 158*q* the nominal dimensions of the hole allow it to be brought out beyond the wall limits, but the tool may cut into the wall due to production deviations (especially if the wall is rough).

The tool will cut properly if the hole is removed from the wall to a distance k (Fig. 158*r*) sufficient to prevent cutting into the wall whatever its dimensional variations.

If this is not possible the hole then should be arranged in a boss (Fig. 158*s*).

4.20. Elimination of Deformations Caused by Cutting Tools

To obtain the required accuracy of machined surfaces, the first condition to be met is their sufficient and uniform rigidity. Otherwise, the less rigid portions are liable to sag under the action of the cutting force and will regain their former position after the cutting is done. This impairs the dimensional accuracy.

The requirement for uniform rigidity is especially important with the present-day highly productive machining methods involving increased cutting forces.

Figure 159*a* shows an erroneous design of a housing with a bracket machined on the upper surface *m*. The cutting force deflects the bracket down (Fig. 159*b*) which straightens after machining (Fig. 159*c*);

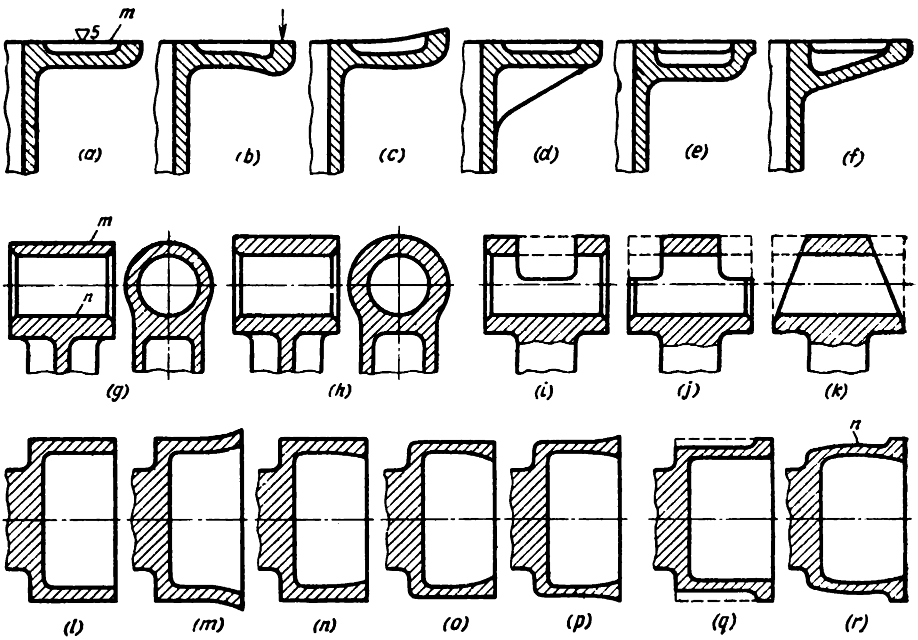


Fig. 159. Elimination of deformations caused by cutting tools

the straightness of the surface is impaired. When the bracket bends too much the resulting vibrations do not allow a well finished surface to be obtained.

In the design shown in Fig. 159*d* ribbing increases the rigidity of the bracket. If outer ribs are not allowed for size reasons the rigidity can be improved by increasing the height of the bracket walls and using internal ribbing (Fig. 159*e*), or else by inclining the bracket walls (Fig. 159*f*).

A wrong rod head design is illustrated in Fig. 159*g*: the nonuniform rigidity of the walls at sections *m* and *n* makes the hole deflect when boring towards the weaker wall and the hole becomes oval. An accurate hole can only be obtained by removing very fine chips, for example with a diamond-tipped tool operating at a fine feed and small depth of cut.

In the design shown in Fig. 159*h* the walls of the head are made thicker to reduce their deformation during machining.

It is practically impossible to obtain accurate holes in parts with local recesses (Fig. 159*i*, *j*) or tapers (Fig. 159*k*). The machining tool knocks against the recessed sections forming steps at the points of transition into the full profile. When using reamers and broaches, these tools deflect towards the weaker wall. The machined walls regain their initial position making the hole oval. The following method is possible: the hole is first finish machined and then the recesses are milled (dashed lines in Fig. 159*i*-*k*). But in this case, too, the walls of the hole are slightly deformed during milling and the hole cylindricity is impaired.

Figure 159*l* shows a hole machined in a cup-shaped part. If the hole is machined first, the force applied by the cutting tool will cause the cup section of minimum rigidity (at its end) to spread out (Fig. 159*m*). After the machining is over the cup walls return to their original position and the blank assumes the shape shown in Fig. 159*n*.

Further external machining deforms the walls in the opposite direction (Fig. 159*o*). The machined part takes the shape shown in Fig. 159*p*. The external and internal surfaces are no longer cylindrical.

The same occurs when the order of machining is reversed, i.e., when the external surface is machined first, and the internal surface next.

The annular rib provided at the cup end for rigidity (Fig. 159*q*, *r*) improves the design. However, in this case, too, the shape may be distorted, if the cup is very long. If the internal surface is machined first, the hole will be accurate enough due to the increased wall rigidity (Fig. 159*q*). During subsequent outside machining (Fig. 159*r*) the cutting force causes the cup walls over the nonrigid portion *n* to deflect inside. After machining the deflected walls diverge and the part becomes barrel shaped.

This can be prevented by providing for another stiff rib at the section *n*, or by making the walls thicker over the entire cup length.

In practice, the accuracy of manufacture is appreciably affected by the rigidity of the cutting tool, operating members of the machine and fixtures used to clamp the blank. Distortions of this nature are eliminated by increasing the rigidity of the tools, proper clamping of the blank, etc.

4.21. Joint Machining of Parts of Different Hardness

The joint machining of parts made from materials of different hardness should as far as possible be avoided.

It is practically impossible to fasten a steel bearing bushing in an aluminium alloy housing with a screw entering partly into the bushing and partly into the housing (Fig. 160*a*), because when drilling is done along the joint line between two such elements the drill deflects towards the softer metal. In this case the fastening should be such as will allow the housing and bushing to be drilled separately (Fig. 160*b*, *c*).

If an aluminium alloy bushing and a steel shaft are drilled together (Fig. 160*d*), the drill will inevitably deflect towards the bushing. It is better to secure the bushing with a central pin (Fig. 160*e*).

Figure 160*f* shows a steel bearing cap attached to a housing made of an aluminium alloy. It is difficult to jointly bore or ream the bearing seats in the housing and cap because the metals differ in

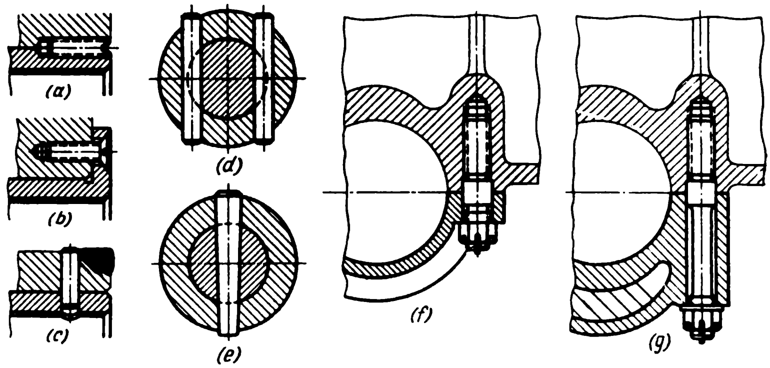


Fig. 160. Machining of parts of different hardness

hardness. The hole deflects towards the softer metal. At the joint between the soft and hard metal the tool operates with shocks and is rapidly blunted. It is impossible to obtain a well finished and accurate surface at the transition portion.

For correct machining the cap should also be made of an aluminium alloy (Fig. 160*g*).

4.22. Shockless Operation of Cutting Tools

During operation the tool should always be kept in contact with the metal. Local recesses, cavities and other irregularities on work surfaces, which hamper the continuous cutting process, should be avoided. As it leaves the work surface the tool is elastically forced towards the recess, and pushed back by the next projection. In these conditions it is difficult to obtain a well finished and smooth surface. A tool subjected to periodic impacts rapidly wears out.

The ribbed bushing design (Fig. 161*a*) is irrational. The tool periodically strikes the ribs and they should therefore be arranged below the cylindrical surfaces being turned (Fig. 161*b*).

When turning flanges with projecting (Fig. 161*c*) or raised (Fig. 161*d*) bosses, and also shaped flanges (Fig. 161*e*) the tool experiences impacts. It is better to make turned flanges round (Fig. 161*f*).

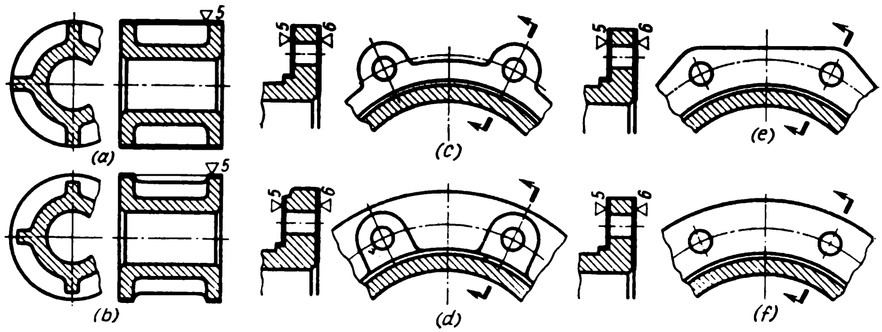


Fig. 161. Shockless operation of cutting tools

4.23. Machining of Holes

It is good practice to make unimportant holes with a surface finish of up to class 5 and a diameter of up to 40 mm by drilling only, without additional machining, leaving the bottom conical (Fig. 162*b*, *e*). The shapes of holes in Fig. 162*a*, *c* and *d*, which require additional machining, are inadvisable.

The operations of preliminary drilling and the features of the finishing tools must be considered when holes are to be machined to a higher grade of accuracy (by counterboring, boring or reaming).

A hole with a flat bottom (Fig. 162*f*) cannot be counterbored or reamed. The cutting cone of the counterbore leaves an unmachined layer of metal in the section *m*.

In the design shown in Fig. 162*g* the hole is drilled first, but the drilling depth is insufficient and an unmachined layer of metal remains in the section *n* after counterboring.

In the correct design in Fig. 162*h* the bore is sunk into the hole bottom to a depth l enough for overtravel of the drill cutting cone, which makes it possible to maintain the specified length l' of finish machining. The drilling diameter is determined by the amount of allowance s for the finish machining.

The same rule should be observed for holes with an undercut groove for tool overtravel. In designs where the drill does not reach the bottom of the hole (Fig. 162*i*) there remains an unmachined layer t which has to be removed when the undercut is bored out. In the advisable design (Fig. 162*j*) the bore is deeper than the bottom of the undercut and the machining of the latter is much easier.

Undercut grooves *m* (Fig. 162*k*) should be avoided in small-diameter holes ($<$ 15-20 mm).

It is practically impossible to ream the hole shown in Fig. 162*l* due to the presence of the cutting cone on the reamer. The bore should be deepened to a distance l (Fig. 162*m*) enough for overtravel of the reamer cone.

Figure 162*n*, *o* shows wrong and Fig. 162*p*, correct designs of threaded holes. The minimum distance l between the hole bottom and finished threads with a full profile is determined by the length

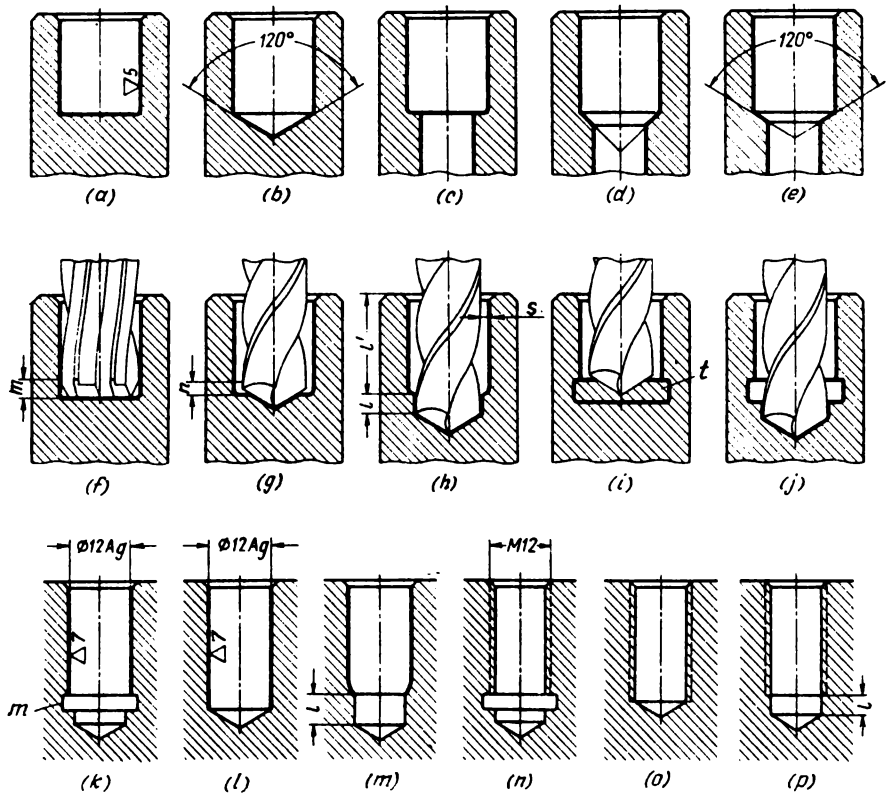


Fig. 162. Machining of holes

of the tap starting section. In finishing taps the length of the starting section is, on the average, $l = (0.3 \text{ to } 0.4)\ d$ where d is the thread diameter.

It is bad practice to drill holes at an angle $\alpha < 70°$ to the surface (Fig. 163*a*). This method requires preliminary drilling (Fig. 163*b*) or milling (Fig. 163*c*) of the hole entrance portion, which complicates manufacture. Machining will be easier if the hole is arranged at an angle larger than 70° to the surface (Fig. 163*d*).

It is better to drill a hole at right angles. Some methods of straightening out the work elements for skew bores in cast parts (Fig. 163*e*) are illustrated in Fig. 163*f-h*.

Examples of wrong and correct arrangement of holes are given in Fig. 163*i*, *j* (pinning a handwheel) and in Fig. 163*k-m* (pinning

a cylindrical part on a shaft). The designs in Fig. 163*j*, *l*, *m* are correct.

Figure 163*n-p* presents methods of drilling holes in a crankshaft, the holes being intended to feed oil from the main journal to the

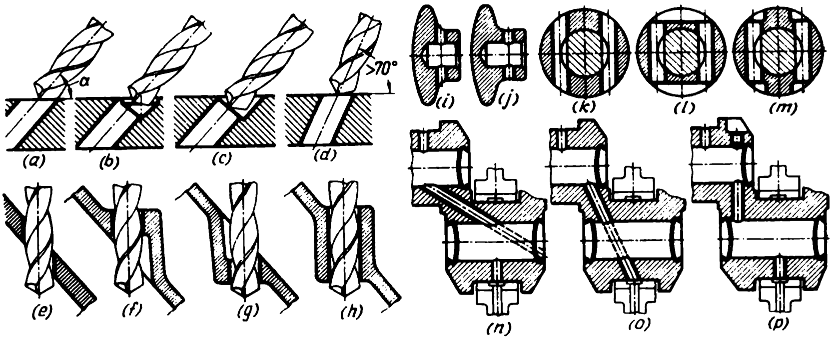


Fig. 163. Drilling of skew holes

crankpin. Most rational is the design with a straight hole through the web (Fig. 163*p*).

Holes obtained by means of ordinary helical drills should never be more than 6-8 diameters deep for otherwise the hole may be misaligned and the drills broken.

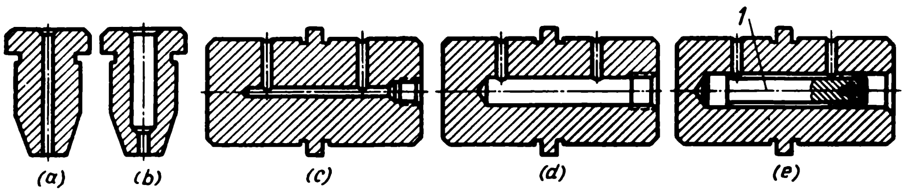


Fig. 164. Drilling of deep holes

It is advisable to reduce the drilling depth to the minimum permitted by the design. Long and thin bores (Fig. 164*a*) should be replaced by stepped ones (Fig. 164*b*).

The long and narrow oil duct (Fig. 164*c*) connecting the bores in the shaft is not just as good as the duct of a larger diameter (Fig. 164*d*). If the cross-section of the duct has to be reduced (for example, for faster oil feed during starting), this can be done by means of insert *1* (Fig. 164*e*).

4.24. Reduction of the Range of Cutting Tools

The range of cutting tools can be reduced if the diameters of accurate surfaces are unified. This is especially important for holes machined by such tools as drills, counterbores, reamers and broaches.

One and the same tool is preferred for the maximum number of operations so that time is not lost in resetting and replacement.

It is good practice to make the transitions between steps and shoulders on turned shafts, which do not serve as bearing surfaces (Fig. 165*a*, *c*), tapered at an angle equal to the plan approach angle

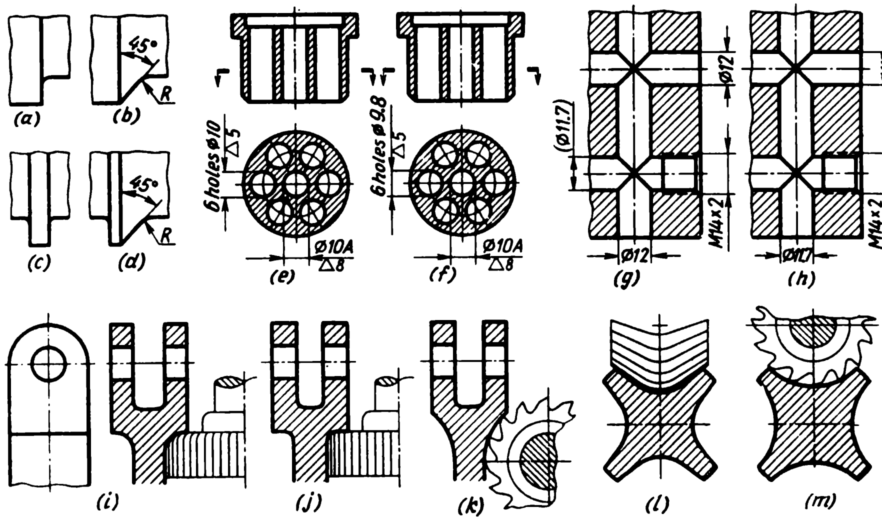


Fig. 165. Reduction of the range of cutting tools

of the cutting edge of a turning tool (usually 45°) and with a fillet at the base equal to the standard tool top rounding $R = 1$ mm (Fig. 165*b*, *d*). This makes it unnecessary to change the cutting tool and undercut the step ends.

Figure 165*e* shows a valve seat with a centre hole of diameter 10*A* for the rod of the valve and with six holes 10 mm in diameter for the passage of working fluid. Two drills are needed to make the holes: one with a diameter of 9.8 mm for a rough machining of the centre hole with a reaming allowance and the other with a diameter of 10 mm to drill the peripheral holes. Only one drill may be used if the peripheral holes have a diameter of 9.8 mm (Fig. 165*f*).

Figure 165*g* illustrates methods of drilling oil ducts in a housing. One of the ducts, stopped with a plug having a thread M14 × 2, is made by a drill with a diameter of 11.7 mm to leave some metal for the thread.

The adjacent ducts have a diameter of 12 mm. In this case it is expedient to machine all the ducts with the 11.7 mm drill (Fig. 165*h*) used to drill the threaded hole.

Special tools are not recommended for piece and small-lot production.

In the forked lever design (Fig. 165*i*) the transition portion between the rod and the fork should be milled with a special radiused cutter. The transition in Fig. 165*j* can be machined with a standard plain milling cutter. The best design is the one with smooth transition portions between the rod and the fork machined with a standard milling cutter (Fig. 165*k*).

Figure 165*l*, *m* (cross-shaped part) shows how form milling can be replaced by plain milling if the shape of the space to be cut out is changed.

4.25. Centre Holes

Parts intended for machining on circular grinding machines or lathes, where the blank is mounted either between centres or in a chuck, with the free blank end being supported by the tailstock centre, are provided with centre holes.

Standard types and sizes of centre holes (according to the USSR State Standard ГОСТ 14034-68) are shown in Fig. 166. Centre holes

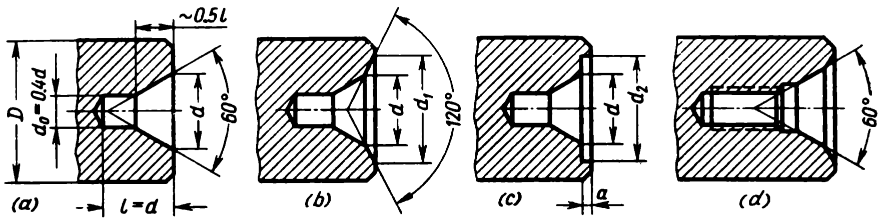


Fig. 166. Centre holes

with a chamfer (Fig. 166*b*) or recess (Fig. 166*c*) which protect the centring cone against dents are used when a part is mounted between centres during tests and also when it is necessary to keep the centres intact in case of returning or regrinding during repairs. Centres with a threaded hole (Fig. 166*d*) are used when a bolt has to be fitted in, and also (for heavy shafts) as a means for lifting the shaft.

The main parameter of a centre hole is the outer diameter d of the cone equal, according to the USSR State Standard ГОСТ 14034-68, to 2.5, 4, 5, 6, 7.5, 10, 12.5, 15, 20 and 30 mm.

Diameter d_1 of the protective chamfer (Fig. 166*b*) is made equal to (1.3 to 1.4) d and diameter d_2 of the protective recess (Fig. 166*c*), to $1.3d$. The depth of the recess a is equal to (0.1 to 0.15) d (the lower limit for holes of large diameter, and the upper one, for those of small diameter).

The working surfaces of centre holes are made to a finish of class 9-10.

A blank can be installed between centres much more accurately and reliably if the maximum size of the centre hole, allowed by the

design of the part, is used. The more massive and longer the part, the larger should the diameter of the centre hole be. The relation $d \approx 0.5D$ (D — shaft diameter) is preferred for the centre holes shown in Fig. 166*a*, and $d \approx 0.4D$, for centre holes with a protective chamfer or recess (Fig. 166*b*, *c*).

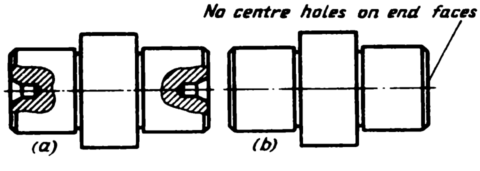


Fig. 167. Centre holes

Centre holes are, as a rule, depicted on drawings as shown in Fig. 167*a* and designated according to standards. The absence of centres on a drawing (Fig. 167*b*) means that the part is machined without mounting it between centres (turning with the part fastened in a chuck, centreless grinding, etc.) or that centres cannot be permitted by the functional purpose of the part. In this case a corresponding inscription should be made on the drawing to prevent a mistake being made (Fig. 167*b*).

Centre holes can be removed by cutting off the centred ends of the shaft. The result is a greater waste of metal and surplus machining. Therefore this method is only used when absolutely necessary.

Centre holes often predetermine the design shape of parts. Such cases are illustrated in Fig. 168*a*, *b* (curvilinear lever), Fig. 168*c*, *d* (bolt with an asymmetric head), and Fig. 168*e*, *f* (part with three journals).

The centring surfaces in hollow shafts are made in the form of chamfers with a central angle of 60°. The choice of manufacturing

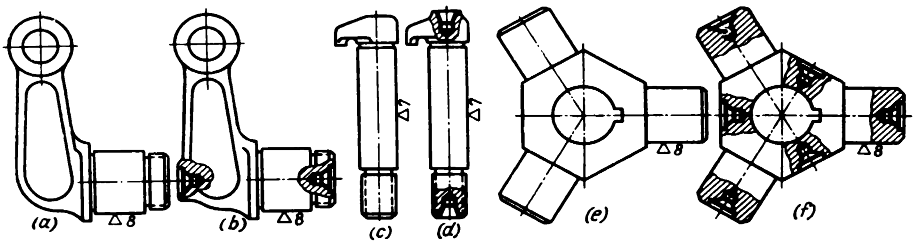


Fig. 168. Centre holes in asymmetric parts

operations can be broadened, the weight of parts reduced and their shape approximated to the form of a body of equal resistance to bending, if the ends of the holes of hollow cylindrical parts are made in all cases with a conical chamfer having a central angle of 60° (Fig. 169*b*) instead of the usual chamfer with an angle of 45° (Fig. 169*a*). If a part is machined in the centres the surfaces of the

centre chamfers are machined to the required finish and provided with protective chamfers or recesses (Fig. 169*c-f*).

Centre chamfers should never be made on interrupted surfaces, for example on shafts with end slots (Fig. 170*a*) and splines (Fig. 170*b*).

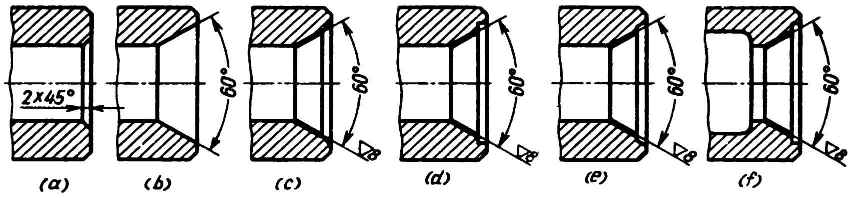


Fig. 169. Centre chamfers

A centre chamfer should be removed to a distance enough for the centre to pass (Fig. 170*c*). When the hole is large in size and stubbed centres may be employed (Fig. 170*d*) this limitation may be disregarded.

A thread should never impinge on the centre chamfer (Fig. 170*e*). If the first threads are crushed in screwing in and out the centring

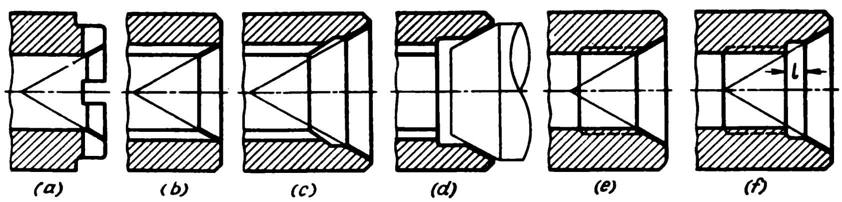


Fig. 170. Shapes of centre chamfers

surface will be damaged and the centre chamfer cannot be used for the second time. The threaded portion should be separated from the chamfer by a recess (Fig. 170*f*) of length l enough for the passage of the centre.

4.26. Measurement Datum Surfaces

These surfaces are usually existing designed elements but sometimes special measuring features have to be introduced.

It is difficult to measure the major diameter D of the cone of a tapered plug (Fig. 171*a*) because of its sharp edge. It is practically impossible to measure the minor diameter d of the part. Parts shaped so can only be measured with the aid of a taper ring gauge.

To make measurement easier it is more practical to provide the major diameter of the cone with a cylindrical belt with a width of b = 2-3 mm (Fig. 171*b*).

In a spherical part (Fig. 171*c*) it is difficult to measure the diameter D of the spherical surface because of its sharp edge. In the sound

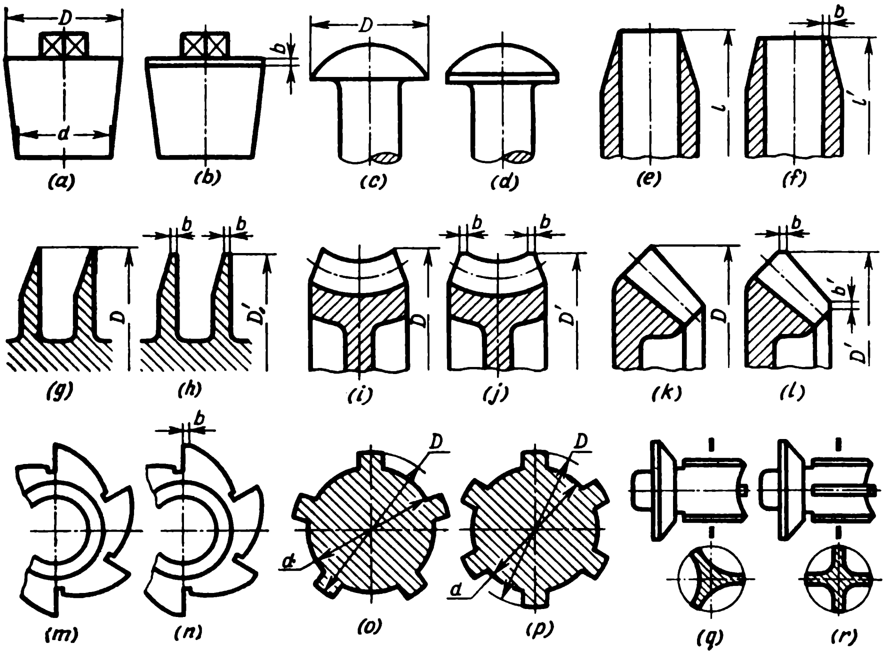


Fig. 171. Measurement datum surfaces

design (Fig. 171*d*) the edge is cylindrical. In addition to making the measurements easier this design of heat-treated parts prevents overheating of the edge.

It is difficult to ensure the proper axial dimension l due to the sharp edges on the end of the tapered part (Fig. 171*e*). The flat section on the end (Fig. 171*f*) makes manufacture and measurement easier.

A wrong annular rib design is shown in Fig. 171*g*, and a correct one in Fig. 171*h*.

It is good practice to provide cylindrical sections of width b (Fig. 171*j*) on the toothed rims of worm wheels (Fig. 171*i*) which facilitate measurement, simplify the axial assembly of the worm drive and prevent concentration stress on the edges of the teeth.

The cylindrical sections b on the teeth of bevel gear wheels (Fig. 171*k*, *l*) form a measuring datum surface and prevent stress

concentration on the top of the tooth. The sections b' make axial mounting of the wheel easier.

Figure 171*m*, *n* shows an example of cylindrical datum surfaces being provided in the design of a ratchet wheel.

Parts with splines can be measured much easier if the number of splines is even. The outer diameter D of a spline shaft with an odd number of splines (Fig. 171*o*) can be measured only by the ring gauge; it is still more difficult to measure the internal diameter d. In the design with an even number of splines (Fig. 171*p*) the diameters D and d can be measured with all-purpose measuring tools.

Figure 171*r* (shank of a tapered valve) shows the design with an even number of centring ribs which is more advantageous than the design in Fig. 171*q* with an odd number of ribs.

4.27. Increasing the Efficiency of Machining

Machining efficiency will undoubtedly increase if the maximum number of surfaces are processed on one and the same machine-tool, at one setting, in one operation with one tool utilizing all the possibilities of the machine on which the main operation is carried out.

In the design of a cylindrical shaft with an eye (Fig. 172*a*) the shaft and the adjacent end of the eye K are machined on a lathe. The surface m is milled to a templet.

In design *b* the eye has a cylindrical form, and in design *c* the eye is spherical. All machining operations (except for drilling the hole and milling the faces n) are performed on a lathe, which appreciably increases the efficiency of machining.

Figure 172*d* illustrates the shoe of friction clutch whose external surface p is to be turned. The fastening flange is of a rectangular shape and requires additional complicated milling operations.

In the rational design *e* the flange is cylindrical, and the entire part is machined on a lathe as an annular blank which is then cut into sectors. To reduce waste the length of the sectors should be such as to accommodate them a whole number of times in the circumference of the blank including the slitting saw thickness.

In the flanged shaft with a square flange (Fig. 172*f*) the side faces of the square are milled to a templet. The shaft with a cylindrical flange (Fig. 172*g*) is machined wholly on a lathe.

The number of resets should be reduced to the minimum on each machine tool so that the maximum possible number of surfaces can be machined in one setting.

Figure 172*h* presents an adapter with two centring bores of different diameter and two rows of offset fastening holes. A slight design change (Fig. 172*i*) makes it possible to through-pass machine the centring bores and fastening holes simultaneously.

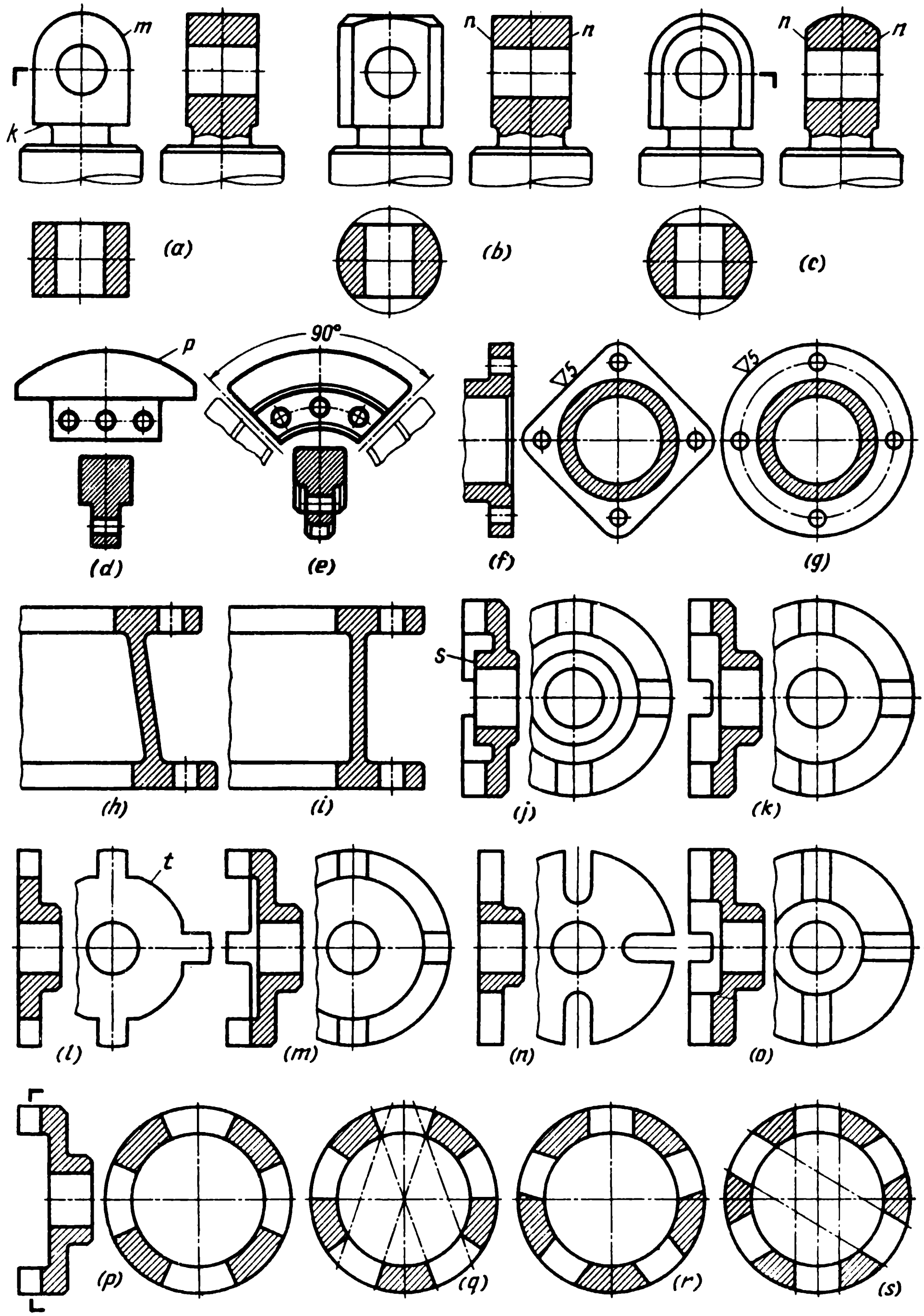


Fig. 172. Increasing of the machining efficiency

The design of the slotted washer (Fig. 172*j*) is poor. The hub *s* protruding into the washer hampers a through-pass machining of the slots which in this instance can be machined only by an unproductive slotting operation.

In the rational design *k* the slots are through milled.

In a four-jaw driver with radial jaws (Fig. 172*l*) the side faces of the jaws are milled in four settings, the blank being each time rotated through 90°. The surfaces *t* between the jaws are planed or milled to a templet.

In design in Fig. 172*m* the radial jaws are replaced by side ones milled in two settings. At each setting two jaws are machined simultaneously. The working faces of each pair of jaws are through-pass machined and the accuracy of the arrangement of the jaws is therefore increased.

The same advantage can be derived if radial slots (Fig. 172*n*) are replaced by side ones (Fig. 172*o*).

The number of slots and their layout should agree with the conditions required by through-pass machining allowing the maximum number of surfaces to be machined at the same time.

If the faces of slots are located radially the number of slots should preferably be *uneven* (Fig. 172*q*). This makes it possible to through-pass machine two opposite faces simultaneously (dash-and-dot lines). When the number of slots is even (Fig. 172*p*) machining is inconvenient and non-productive.

Conversely, in the case of straight-sided slots through-pass machining requires an *even* number of slots (Fig. 172*s*). Machining is difficult when the number of slots is uneven (Fig. 172*r*).

Machining at an angle to datum surfaces should be avoided. This complicates setting up of the machine-tool because the product has to be mounted on swivel tables or attachments.

Figure 173*a*, *c* shows examples of unsound arrangement of holes in frames. Machining is considerably simplified if the holes are parallel (Fig. 173*b*) or normal (Fig. 173*d*) to the datum surfaces.

In the design *e* of an eye (Fig. 173) the threaded hole for an oiler is positioned at an angle, which means that a jig is necessary for drilling the hole. In design *f* the hole is positioned on the axis, and can be drilled and threaded when the eye is turned on a lathe.

In the design *g* in Fig. 173 of a sealing unit the inclined drain hole *m* can be made parallel to the shaft axis if a slot *n* is milled in the seal cover (Fig. 173*h*) or if the diameter of the cover recess (Fig. 173*i*) is increased to $D = 2h + d$ (h is the distance of the drain hole to the shaft centre and d the drill diameter).

In the impeller of a centrifugal machine (Fig. 173*j*) the thickening of the impeller disk towards the hub required for better strength can be attained if the surfaces *s* between the blades are inclined. This makes it necessary when milling for the impeller to be held

in a fixture on a canted centring pin. In the design *k* in Fig. 173 the disk can be thickened towards the hub if the back surface *t*

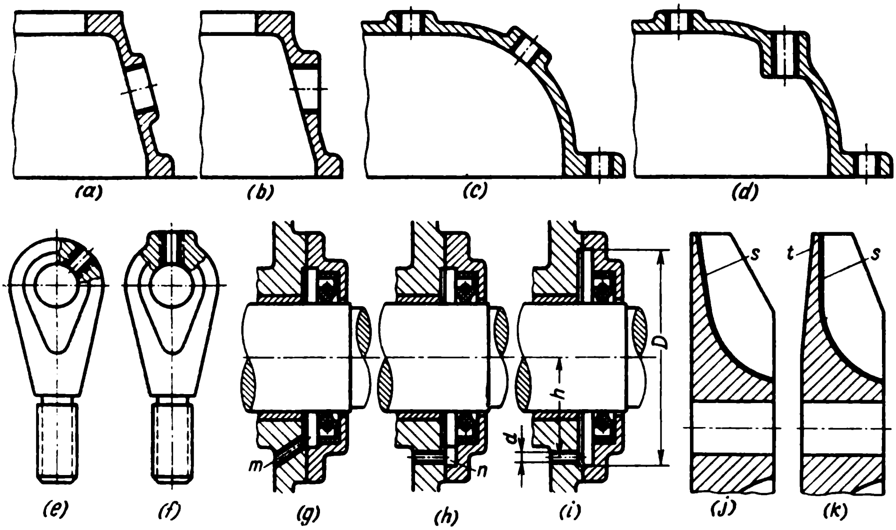


Fig. 173. Elimination of machining at an angle

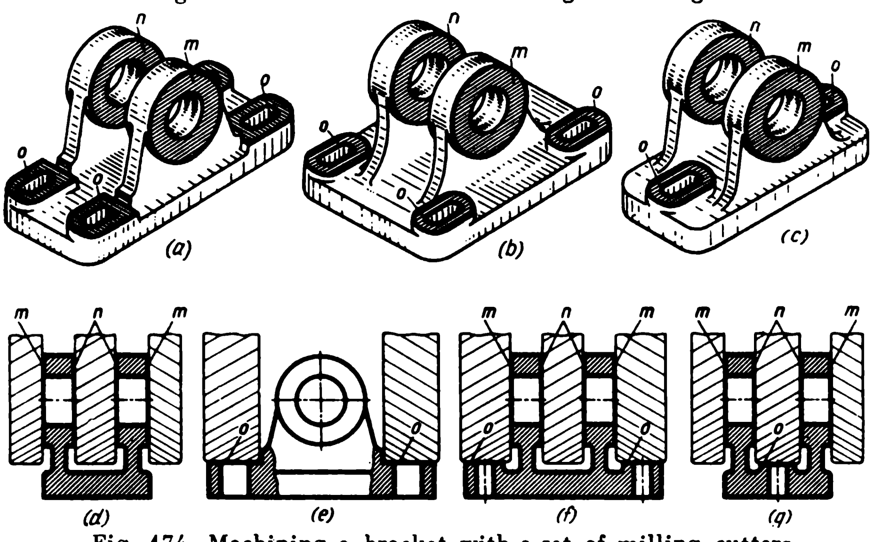


Fig. 174. Machining a bracket with a set of milling cutters

of the impeller machined by turning is slightly tapered. The surfaces *s* between the blades are milled.

Machining productivity can appreciably be increased by the use of combination tools which simultaneously machine several surfaces (core drills, block cutters, sets of milling cutters, etc.).

The bracket (Fig. 174*a*) processed over the external *m* and internal *n* side faces of the eyes and also over the surfaces *o* of the fastening bosses is machined with a set of plain milling cutters in two settings. The first setting is used to machine the side faces *m* and *n* of the eyes with a set of three milling cutters (Fig. 174*d*). Then, the part is swivelled through 90° and the boss surfaces *o* are milled with a set of two cutters (Fig. 174*e*).

Dislocation of the bosses in relation to the eyes (Fig. 174*b*) allows the part to be machined in a single setting with three milling cutters. The cutter side faces (Fig. 174*f*) cut the surfaces *m* and *n* of the eyes, and the peripheries of the two outer cutters process the surfaces *o* of the bosses at the same time.

In the very compact design *c*, the fastening bosses are arranged between the eyes and are machined by the periphery of the internal cutter (Fig. 174*g*) at the same time as the internal side faces *n*.

4.28. Multiple Machining

In large lot and mass production, the tendency is to machine parts in groups to a preset operation with establishment of the blanks in quick-acting machining fixtures.

Consecutive machining (Fig. 175*a*) reduces handling time (the time needed to mount the blank and adjust the machine tool).

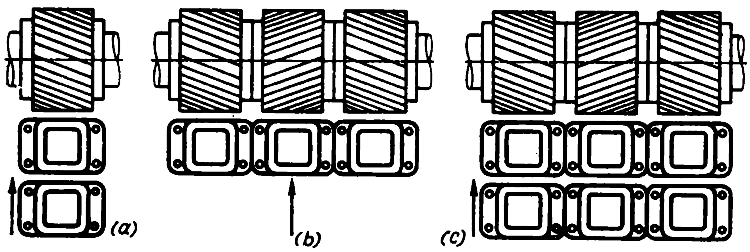


Fig. 175. Diagrams of group machining

Parallel machining (Fig. 175*b*) reduces machining time in proportion to the number of blanks being simultaneously machined.

Parallel-consecutive machining (Fig. 175*c*) is the most productive.

For all these methods through-pass machining is obligatory.

Figure 176*a* illustrates a circular nut with radial wrench slots which are located below the thread by the amount *m*. The slots are machined by non-productive indexing method (only by planing or slotting). The shape of the part does not permit milling.

In the design *b* in Fig. 176 the slots are milled, but as in the previous case the part cannot be group machined. If the slots are located higher in relation to the thread by the amount *n* (Fig. 176*c*) a number of nuts mounted on a mandril can be consecutively machined together in groups by the generation method with the aid of a hob.

The lug (Fig. 176*d*) with a slot profiled to a circumferential arc is suitable only for piece machining. A straight slot (Fig. 176*e*) permits consecutive group through-pass machining.

Figure 176*f* shows plates *1* and *2* clamped by distance bolts *3*. The bolts can be turned only individually. The manufacture of the

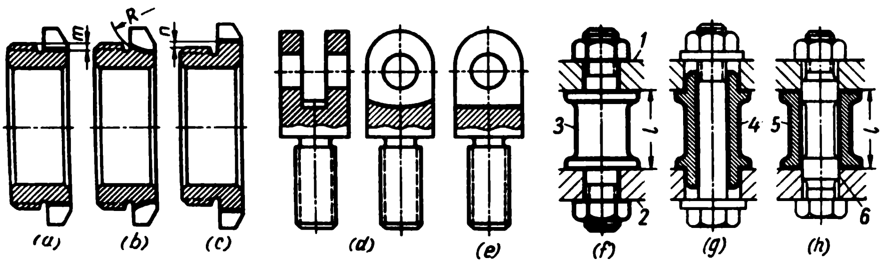


Fig. 176. Examples of group machining

bolts is complicated because an accurate distance *l* has to be maintained between the shoulders.

In the design *g* in Fig. 176 the plates are tightened up against bushing *4*. The centring shoulders make group machining of the bushings impossible.

In the design *h* in Fig. 176 the distance bushing *5* has flat end-faces and the plates and bushings are mutually centred by means of dowel bolts *6*. In this design the distance *l* between locating surfaces of the bushings can easily be maintained by machining the bushings in groups on a surface grinding machine, the bushings being clamped on a magnetic chuck. Bushings can be machined much more quickly on a rotary table grinding machine.

Parts intended for consecutive and parallel-consecutive group machining should have datum surfaces that will ensure their correct mutual positioning during machining. When milling, datum surfaces may be the bases of the parts and their side faces. When cylindrical parts are machined, the datum surfaces are usually centre holes. The parts are mounted on a mandril and machined in a group.

The sections of workpieces intended for machining should be durable enough to withstand deformation under the action of the cutting forces.

Gears in which hub faces protrude in relation to rim faces (Fig. 177*a*) are not suitable for group machining as the gear rims are not secured

rigidly during machining and can deform and vibrate under the cutting force.

It is preferable to make hubs flush (Fig. 177*b*) or with a small (0.1-0.2 mm) clearance *s* (Fig. 177*c*) in relation to the rim.

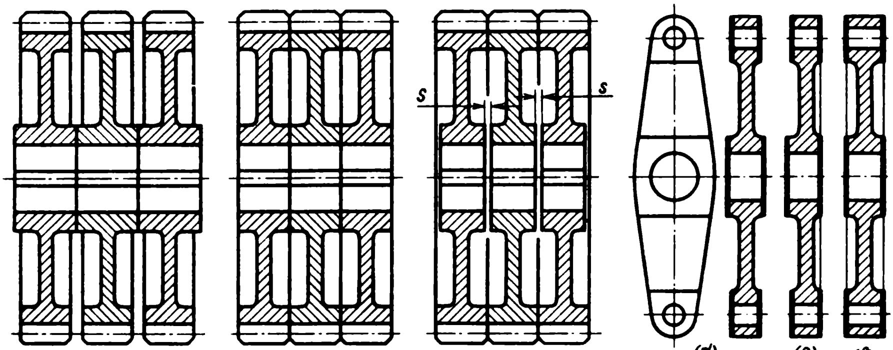


Fig. 177. Elimination of deformation of blanks in group machining

It is good practice to clamp blanks using not the hubs but special end disks resting against the rims.

Figure 177*d-f* shows a lever requiring milling over its external contour. The protruding hub faces (Fig. 177*d*) do not allow the set to be clamped tightly. Design *e* allowing the parts to be clamped in pairs is better but the best design *f* for group machining has all faces arranged in one plane.

Chapter 5

Welded Joints

In mechanical engineering, welding is extensively employed to manufacture structures from plate rolled stock (reservoirs, tanks, hoppers, coverings, linings, etc.) and from pipes and shaped rolled stock (frame structures, trusses, columns, pillars, etc.). Nowadays housings and base members are also made by welding, including the most massive and stressed parts (for example, the beds of presses and hammers).

To simplify the manufacturing process it is sometimes expedient to separate intricate forgings and castings into simpler elements and connect them by welding (*weld-forged* and *weld-cast* structures).

In individual and small-lot production welded structures are used instead of one-piece forgings when the manufacture of dies is not justified by the scale of production, and also as a means to make the manufacture of complicated parts less expensive. Low-carbon steel ($<$0.25 per cent C), low-alloy steel with a small content of C and nickel steel weld very well. High-carbon, medium- and high-alloy steels are more difficult to weld.

It is difficult to weld nonferrous metals (copper and aluminium alloys) in view of their high heat conduction and easy oxidation (formation of refractory oxide spots), which makes the use of flux necessary.

The strength of welds is inferior to that of solid material because of the cast structure of the welded joints with its dentritic and acicular crystallites typical of cast metal. A coarse crystalline structure forms in the metal adjacent to the weld seam and in the affected zone.

The strength and resilience of the material in a weld are impaired by penetration of slag, formation of pores and gas bubbles and also because of chemical and structural changes in the weld (alloying elements burn-out, formation of carbides, oxides and nitrides).

If the material of a weld is saturated with air nitrogen even in small quantities the weld will lose much of its plasticity (Fig. 178) and will become much more brittle.

Metal contraction during solidification causes internal stresses in the weld and in the adjacent area with possible warping of the product.

The reduction of strength in parts made of low-carbon steel (whose plasticity prevents the appearance of internal stresses) is not large, and is almost immaterial in structures operating under a static load and under moderate stresses. However, this reduction is very

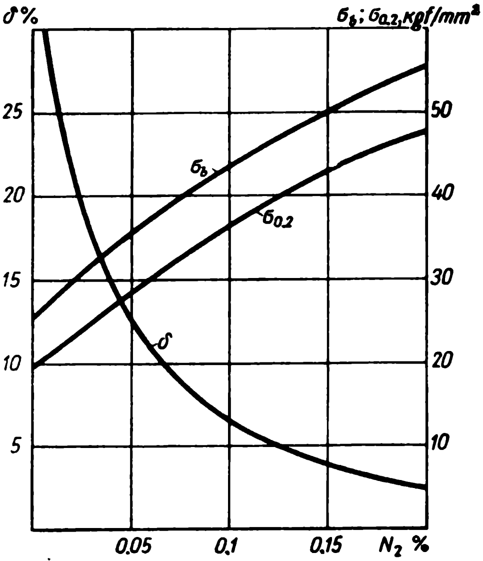


Fig. 178. Effect of nitrogen on the mechanical properties of low-carbon steel

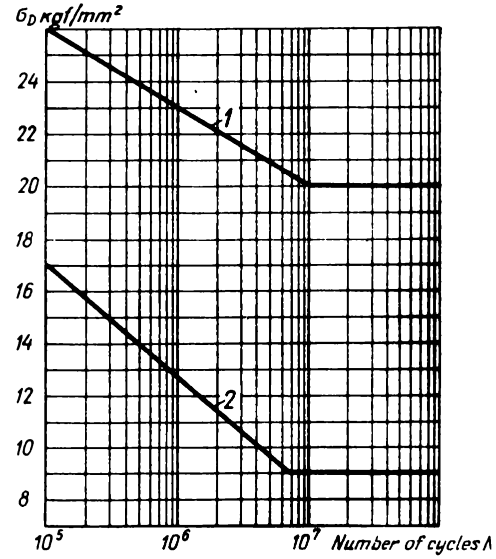


Fig. 179. Fatigue curves
1—solid specimen; *2*—specimen with a circular weld

tangible in structures loaded cyclically, especially if they are made of high-strength steel sensitive to stress concentration.

The effect of welds on cyclic strength is plotted on the diagram in Fig. 179 illustrating the test of a solid cylindrical specimen made of a low-alloy steel (curve *1*) and a specimen of the same steel with a circular V-weld (curve *2*). The presence of the welded joint reduces the fatigue limit more than twice (from 20 to 9 kgf/mm^2). A stress of 15 kgf/mm^2, safe for a solid specimen, is liable to destroy a welded specimen already at 3×10^6 load cycles.

Submerged arc welding or welding in the atmosphere of inert or reducing gases is employed to prevent chemical transformations in the welded metal.

Welding causes *warping* of parts, which is more severe the greater the heat-affected zone (gas welding) and the greater the length and cross section of the welded joints.

Warping can be prevented if a part is welded in rigid holding fixtures and by special methods (intermittent, multilayer or multipass and step and step-back welding). The warping can be removed

Table 4

Principal Welding Methods

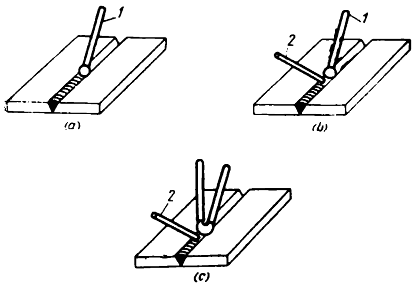
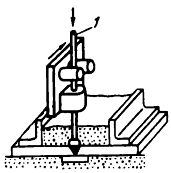
Welding method	Description
Manual electric arc welding 	The most widespread and universal method of welding. It is performed by means of an arc struck between a fusible metal electrode *1* (direct arc) and metal surface. The weld is protected against oxidation by thick-coated electrodes with the first coat liberating liquid slag and reducing gases (CO, H_2) when the arc burns. Welding by carbon electrodes with a direct (*b*) or an indirect (*c*) arc with the rods *2* is mainly reserved for thin-walled parts made of nonferrous alloys. Carbon electrodes are very popular for arc cutting (especially of alloy steels)
Automatic submerged arc welding 	Used in large-scale production to join parts by straight and circular welds. This method implies using bare wire *1* as electrode and the welding is conducted under a layer of flux. The productivity of the process is 5-10 times higher than that of the manual electric arc welding, and the weld has a high quality. Shaped (in plan), short and scattered welds are accomplished by semiautomatic welders in which the welding wire is fed through flexible hoses.

Table 4 (continued)

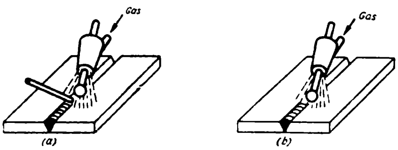
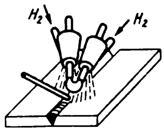
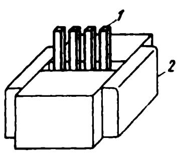
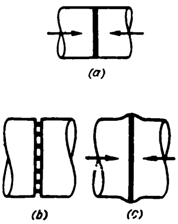
Welding method	Description
Gas-shielded welding 	Welding is done by nonconsumable (*a*) or consumable (tungsten) electrodes (*b*) in a flux of inert gases (argon, helium). The method is used to join parts made of high-alloy steel, titanium, nickel, aluminium and magnesium alloys. Carbon steel is welded with a less expensive carbon dioxide gas.
Atomic hydrogen welding 	Welding is done by an indirect arc with the use of nonçonsumable electrodes in a hydrogen flux which, being an active reducing agent, effectively prevents oxidation of the weld.
Electroslag welding 	Used to connect large blanks (frames of large machines, high-pressure reservoirs). The weld is formed in the clearance between the parts being joined by the fusion of laminated electrodes *1* under a layer of synthetic slag. The outflow of molten metal and slag from the clearance is prevented by water-cooled slide blocks or ceramic linings *2*.
Resistance welding 	*Resistance* butt welding (*a*) is employed to join parts with small cross section. The end-faces of the parts are compressed by a hydraulic press, and the current is switched on to bring the metal in the joint to a plastic state. In the case of *flash* welding the joint is first compressed by a small force and then the current is switched on. This generates a large number of microarcs in the joint which fuse the metal (*b*).

Table 4 (continued)

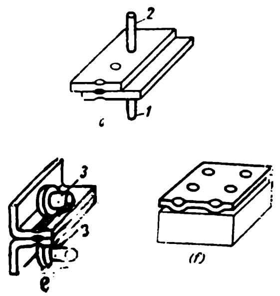
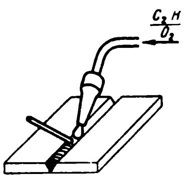
Welding method	Description
 	After fusion the joint is compressed by a hydraulic press (*c*). Flash welding is employed to join parts of large cross section and also parts made of heterogeneous materials. When spot welding is used for lap joints (*d*) the plates are drawn between a stationary *1* and a movable *2* electrodes which periodically compress the plates forming a spot weld. Strong-tight lap joints are formed by seam welding with roller electrodes *3* (*e*). Thin sheets are joined to massive parts by means of projection welding. First flutes are punched on the sheet (*f*). The parts are then compressed between copper electrode plates resulting in fusion and welding of the projections.
Oxyacetylene welding  	Performed in the reducing flame of an injector burner. The addition agent is metal wire or rods similar in composition to the metal of the parts being welded. The quality of the joints is lower than in arc welding. Oxyacetylene welding is predominantly employed to join parts made of carbon steel in small-lot production. Oxyacetylene cutting is applied on a wide scale and noted for its high efficiency and better quality of cutting than electric arc cutting.

Table 4 (continued)

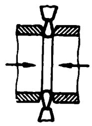
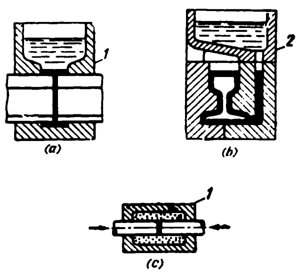
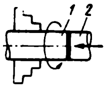
Welding method	Description
Gas-pressure welding 	The edges to be connected are heated by oxyacetylene flame and pressed together by an up-setting mechanism. The method is widely utilized to weld pipelines on site, the joint being heated by burners arranged in a circle.
Thermit welding 	This method is mainly employed to weld structures on site. The source of heat is the exothermic reaction of reduction of iron oxides by aluminium (aluminium thermits). The cleaned joint of the parts being welded together is enclosed in a detachable ceramic mould *1* (*a*) with thermit which is ignited by a phosphorus primer. The reaction produces aluminium oxide that floats up in the form of slag, and molten iron which fills the gap in the joint. Welding is completed after the joint is compressed. An improved method consists in burning the thermit in a separate mould *2* and filling the joint with molten iron (*b*). Power transmission lines are connected by muffle welding with magnesium thermit (mixture of iron oxides with magnesium). The ends of conductors are inserted into muffle *1* (*c*) and are pressed together with a screw clamp.
Friction welding 	Performed by the heat liberated when one of the parts (*1*) being welded is rotated in relation to the other stationary part (*2*) under an axial force. The method is used for butt-welding of small, mainly cylindrical, parts.

Table 4 (continued)

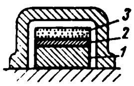
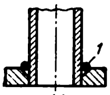
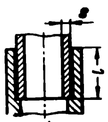
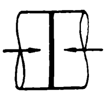
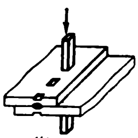
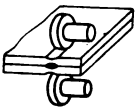
Welding method	Description
Explosion welding  	Used to join thin sheets to massive ones (plating of steel with copper, brass, titanium alloys, etc.). A layer of explosive *3* (ammonite) is placed on the surface of the parts to be welded and is exploded by a detonator. The explosion pressure joins the sheet tightly to the base material.
Furnace welding  (a)  (b)	Used to join parts on cylindrical shoulders (connection of flanges to pipes, or of pipes in frame structures). A bronze or brass ring *1* (*a*) is fitted in the joint, or the joint is greased with a paste of powdered bronze and flux (*b*). Prepared products are heated in an electric furnace in a reducing atmosphere (natural gases) up to a temperature of 1100-1150 °C.
Press cold welding  (a)  (b)  (c)	Used to connect plastic metals (Cu, Ni, Al, Zn, Cd, etc.). The cleaned and degreased joint surfaces (*a*) are compressed by a pressure exceeding the yield point of the material. The surfaces are strongly joined due to the diffusion and recrystallization processes occurring in the compression zone. Lapped sheets are welded under a pressure of round or straight dies (spot welding, *b*) or by roll welding (*c*). Parts made of nonferrous metals (contact points, seats) are welded to steel parts by pressing them into conical seats.

Table 4 (continued)

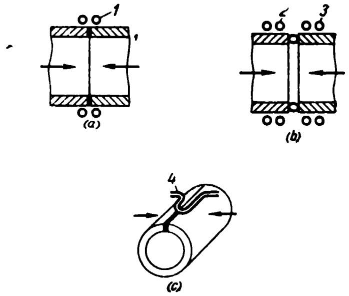
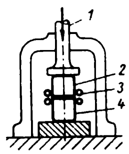
Welding method	Description
Induction welding 	Done by heating the edges to be joined with an inductor *1* (*a*) through which passes a high-frequency current (5-20 kHz), the edges being afterwhrds compressed by an upsetting mechanism. When pipes are welded by the arc resistance method the ends of the pipes are heated by means of opposite directed current in the inductors *2*, *3* (*b*). The currents induced in the joint form a rapidly revolving annular arc which fuses the metal. Welding is completed by compressing the joint. Induction welding is widely applied in automatized pipe production (*c*). A blank rolled into a pipe is drawn through inductor *4* which heats the joint and the pipe edges are compressed.
Diffusion welding 	The joint of parts *2* and *4* being welded is heated by inductor *3* and compressed by ram *1* in a high-vacuum chamber (10^5-10^8 mm Hg) or in atmosphere of inert gases (argon, helium). Heating to 750-800 °C makes a good and reliable joint. This method can be applied to weld refractory and heat-resistant alloys, cermets and ceramics. Currents with a radio-frequency range of 50-200 kHz are employed to weld thin parts made of copper, aluminium and nickel alloys and also stainless steel.

Table 4 (continued)

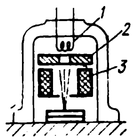
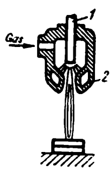
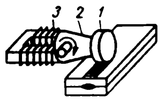
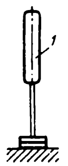
Welding method	Description
Electron-beam welding 	Performed in vacuum by a current of electrons emitted by a tungsten spiral *1* under a high voltage of 250 kV and passed through a circular anode *2*. The current of electrons is focussed by electromagnetic coils *3*. The temperature at the focus point is from 3000 to 10,000 °C; the heating spot ranges from 2-3 mm to several hundredths of a millimetre. This method can be employed to weld parts (with a thickness of several microns) arranged in enclosed spaces (vessels, housings) permeable by electron beams.
Plasma arc welding 	Effected by a jet of an inert gas (nitrogen, helium, argon) ionized by putting it through an electric arc struck between a tungsten electrode *1* and a water-cooled copper nozzle *2*. The temperature along the axis of the jet is 15,000-18,000 °C. In plasmatron welders the gas is ionized by a high-frequency electromagnetic field. The jet of plasma is formed with the aid of electromagnetic coils. The temperature of the jet is up to 40,000 °C. This method can be utilized to weld and cut most refractory materials (including ceramics).

Table 4 (continued)

Welding method	Description
Ultrasonic welding 	This method (with frequency 20-30 kHz) is applied to join nonferrous metals and plastics. The parts are compressed by a vibrating contact jaw *1* connected by a waveguide *2* with a magnetostrictive oscillator *3*. High-frequency oscillations heat the joint and cause diffusive interpenetration of the atoms of the materials being joined. In radio-electronics ultrasonic welding is employed to connect parts up to several microns thick.
Laser welding 	Effected by a concentrated light beam produced by laser *1* (ruby or neodymium crystal). The temperature of the axis of the beam is up to 10,000 °C; the heating spot ranges from several microns to several hundredths of a millimetre. In radio-electronics laser welding is used to connect parts up to several microns thick.

after welding by stabilizing heat treatment (low annealing at 600-650 °C).

The mechanical properties of welded joints depend on the welding process and in manual work on the skill of the welder. Careless welding and improper methods will cause defects impairing the life of the weld and its strength.

In manually welded joints the strength characteristics vary within the weld, the product or a group of the products.

Important welded joints are tested by magnetic, X-ray and gamma-ray methods. The ultrasound test is the most sensitive and accurate.

Large lots of welded products are tested selectively by cutting up of specimens, by tensioning, bending and flattening them and by investigating their microstructure and chemical composition of the metal in the weld. The principal welding methods are illustrated in Table 4.

5.1. Types of Welded Joints

The main types of joints made by arc and gas welding are as follows: butt (C), corner (У), lap (H) and tee (T).

Fillet welds of triangular profile are made *straight* (Fig. 180*a*), *convex* (Fig. 180*b*) and *concave* (Fig. 180*c*). The most common is a straight (*normal*) weld. Convex welds (also called *reinforced* welds) have a tendency to form *undercuts* (poor penetration at the points *m* where the weld adjoins the walls of a part) and possess a lowered cyclic strength. Concave welds are the strongest but their manufacture is more difficult and less productive.

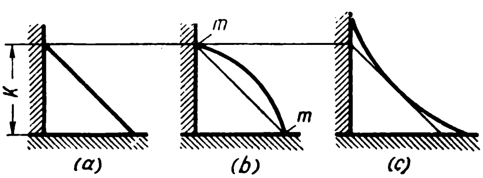


Fig. 180. Fillet welds

The design leg K is the principal dimensional characteristic of fillet welds.

When thin sheets (less than 4 mm) are welded the leg of welds in lap joints is made equal to the thickness s of the sheet (Fig. 181*a*).

For thicker materials (4-16 mm) the leg of a weld can be found from the relation

$$K = 2 + 0.4s \text{ mm} \tag{5}$$

When materials of various thickness (Fig. 181*b*, *c*) are welded the leg is made equal to the thickness s of the thinner material, but not larger than indicated in formula (5). In this case a concave weld is preferred.

In corner joints with the same thickness of the walls (Fig. 181*d*) the length of the leg depends on the thickness of the edges. In corner and tee joints

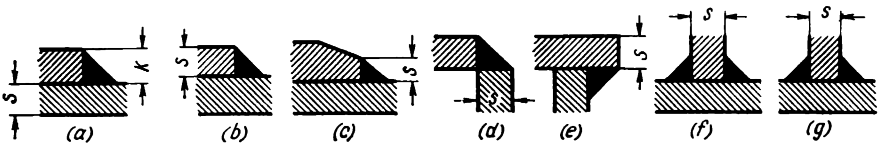


Fig. 181. Dimensions of fillet welds

(Fig. 181*e*, *f*) where the dimensions of a weld may be arbitrary the leg is equal to the thickness s of the elements being welded together, but not larger than the values in formula (5).

When members of various thickness are tee-welded (Fig. 181*g*) the leg is equal to the thickness s of the thinner element. It is preferable to make concave welds.

Lapping is the most simple and reliable method of joining plates (Fig. 182*a*, *b*).

The shortcoming of this method is that lap joints subjected to the action of tensile and compressive forces are bent by a moment

approximately equal to the product of the acting force and the sum of the half-thicknesses of the plates being welded (Fig. 182*a*), and are therefore deformed (Fig. 182*b*). The two welds drastically reduce

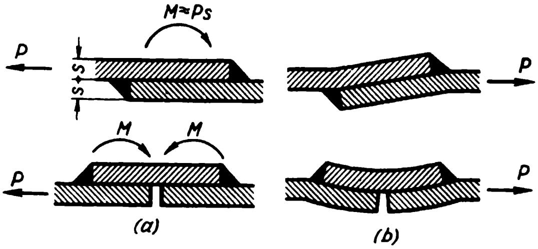


Fig. 182. Operational diagrams of lap joints

the welding productivity, and the weight of the joint is greater than in the case of butt joints.

Lap joints also include *slotted* (*plug*) *welds* formed by fusing up round (Fig. 183*a*) or elongated (Fir. 183*b*) prearranged holes in one of the plates to be connected (these joints are sometimes called *rivet welds*). The laborious manufacture, low strength and poor tightness of the weld make this joint one of the worst which may be employed only when the design requirements do not allow welding by other more productive methods.

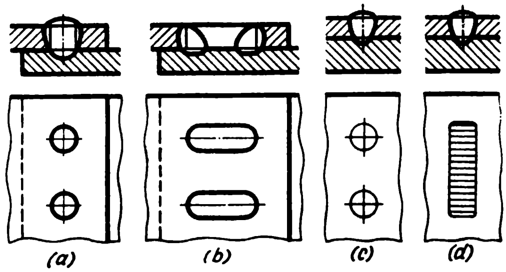


Fig. 183. Slotted (*a*, *b*) and transfusion (*c*, *d*) welds

If one of the members being welded is less than 6-8 mm thick slotted welding is replaced by the simple and effective operation of *spot penetration* (Fig. 183*c*) of the thinner element (*poke welding*) or *seam transfusion welding* (Fig. 183*d*).

When thin (<3 mm) sheets are butt-welded at an angle the edges are flanged (Fig. 184*a*, *i*).

The edges of plates with an average thickness of <8 mm for manual arc welding and <20 mm for automatic welding are made straight (normal to the plane of the plate). For weld penetration through the entire cross section, the parts to be welded are assembled with a clearance m = 1-2 mm (Fig. 184*b*, *j*) filled with molten metal during welding.

In the case of a greater thickness the edges should be prepared mainly by chamfering to produce a weld pool and ensure penetration through the entire cross section.

The principal types of preparation are illustrated in Fig. 184*c-h* (butt joints), *k-m* (corner joints) and *n-p* (tee joints). The sharp corners are broken, leaving belts with a height of h = 2-4 mm (Fig. 184*c*).

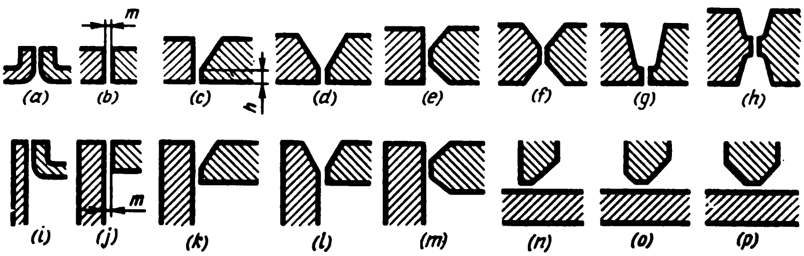


Fig. 184. Preparation of edges

Round chamfers are turned, and straight ones—milled or planed. If the thickness of the edges is over 15-20 mm chamfers are removed by automatic gas cutting.

Preparation with curved bevels (Fig. 184*g*, *h*) is mainly employed for straight and circular welds. A complicated milling operation to a templet is required to prepare edges having an irregular shape in plan.

5.2. Welds as Shown on Drawings

According to Soviet standards, welds are shown on drawings by solid basic lines which coincide with the edges of parts to be welded together. Invisible welds (arranged on the reverse side of the projection) are designated by dash lines.

Welds of spot and seam resistance welding as well as welds obtained by transfusion are shown by dash-and-dot lines drawn through the centres of the welded sections.

A weld is designated by an inclined extended line with an arrow pointing to the line of the weld. The horizontal wing is used for the basic symbol of the weld including:

(1) designation of the kind of welding (Russian letters) (Р—manual, А—automatic, П—semiautomatic);

(2) letter index of the type of welding (Э—arc welding, Г—gas welding, Ф—submerged arc welding, З—gas-shielded welding, Ш—electroslag welding, Кт—resistance welding, Уз—ultrasonic welding, Тр—friction welding, Х—cold welding, Пз-plasma arc welding, Эл—electron-beam welding, Дф—diffusion welding, И—induction welding, Гп—gas-pressure welding, Тм—thermit welding, Лз—laser welding, Вз—explosion welding);

(3) graphical symbol of the type of weld (with the dimensions of the weld when necessary).

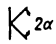
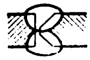
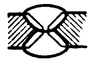
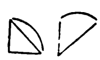
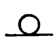
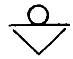
Welds are usually designated on the drawings of welded joints in an abbreviated form. The letters P, A and П are omitted and all the pertaining problems are solved by the process engineer of the welding department depending on the scale of production and available equipment.

The letter (Э) designating electric welding is also omitted because it is the most widespread type of welding. The letters Кт (resistance welding) are not written since the kind of welding is here fully determined by the symbol of the weld. The other letters are given only if a joint should be formed by a certain method of welding.

Thus, most frequently, the designation of a weld consists only of a graphical symbol.

Some symbols are illustrated in Table 5.

Table 5

Type of weld	Symbol	Joint	Type of weld	Symbol	Joint
Fillet weld (*K* — design leg of weld)			Single-V butt weld		
Lap spot weld			Single-V blunted butt weld		
Double-flanged butt weld			Double-bevel butt weld		
Square butt weld			Double-V butt weld		
Single-bevel butt weld			Single-J butt weld		
Single-bevel blunted butt weld			Single-V butt weld		
Convex (reinforced) weld			Remove reinforcement to the surface of edges being welded		
Concave weld			Process the weld to a smooth transition to base metal		

The symbols 4-7 mm high are drawn by thin lines. The angle $\alpha \approx 45^\circ$ and the distance between the adjacent parallel lines of the symbol is not less than 0.8 mm.

The extension lines are drawn as a rule on the visible welds on the projection where the weld is most clear (ordinarily on a plan projection). Extension lines should never be repeated simultaneously on several projections (for example, in plan and in cross section).

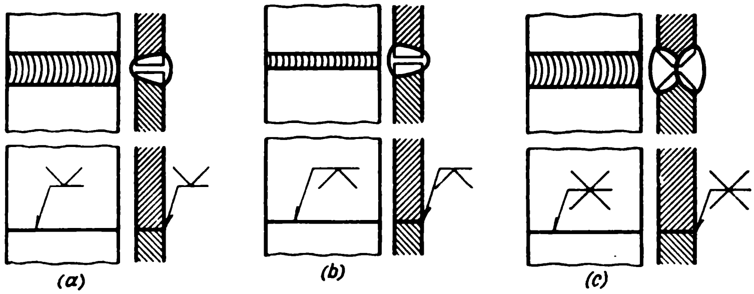


Fig. 185. Designations on extension lines

Symbols are marked above the wing if the extension line is drawn from the face side of the weld (Fig. 185*a*) and under the wing (in an inverted position) if the extension line is drawn from the reverse side of the weld (Fig. 185*b*). The symbols for two-side symmetric welds are written in the middle of the wing (Fig. 185*c*).

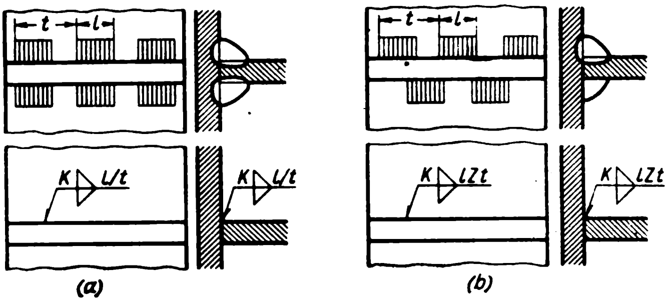


Fig. 186. Designation of intermittent welds

The designations of intermittent welds include the length l and the pitch t of the welded portions (the diameter d and the pitch t of the spots are indicated for spot welds) separated in the case of chain welds by a skew line (Fig. 186*a*) and for staggered welds by the sign Z (Fig. 186*b*).

Designations of welded joints are illustrated in Tables 6-10.

Figure 187 shows some additional symbols. Welds made to a closed contour are designated by a circle at the intersection of the extension line and the wing (Fig. 187*a*). Welds done during assembly are marked by the symbol ⅂ (Fig. 187*b*).

Butt Joints

Type of joint	Weld	Symbol	Depiction of weld on drawings: in plan, face side	Depiction of weld on drawings: in plan, reverse side	Depiction of weld on drawings: in cross section
Single-flanged butt joint					
Double-flanged butt joint					
One-side square butt joint					
Two-side square butt joint					
Square butt joint with detachable strap					
Square butt joint with permanent strap					
Lock joint					
One-side single bevel butt joint					
Two-side single-bevel butt joint					

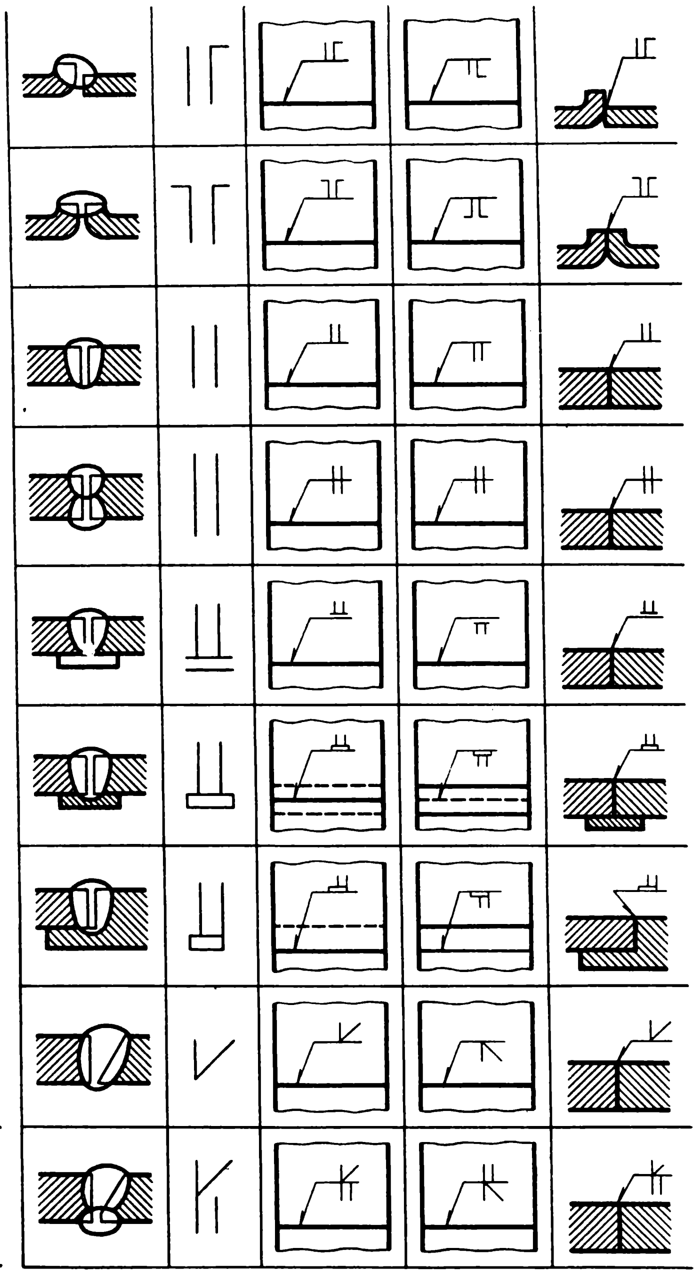
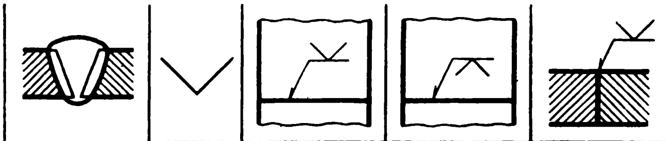
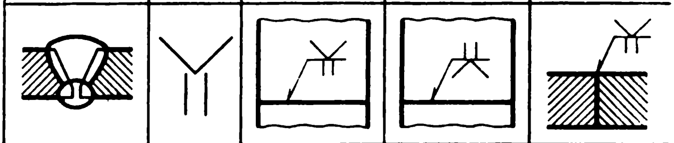
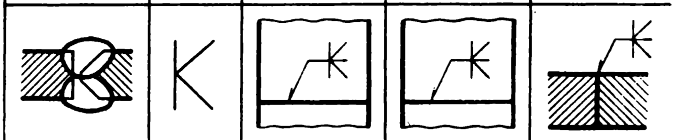
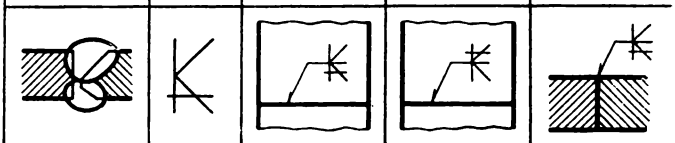
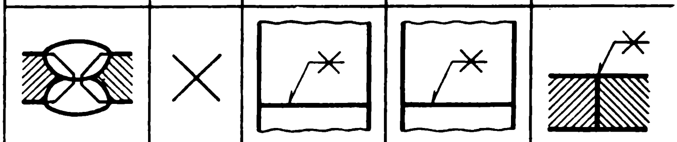
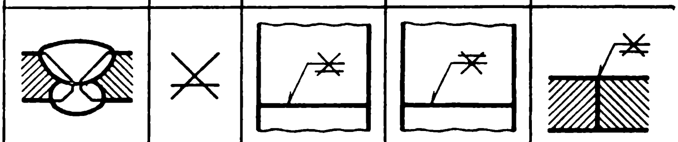
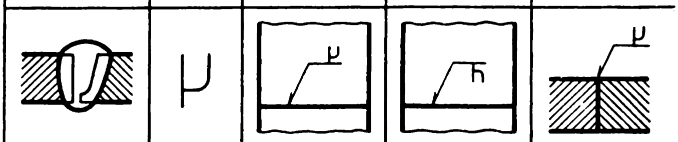
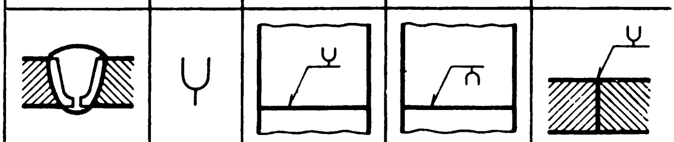
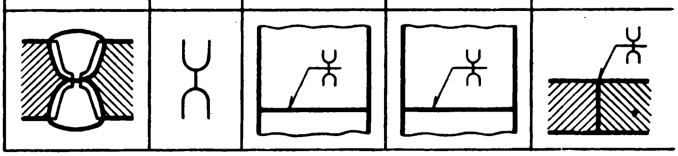


Table 6 (continued)

Type of joint	Weld	Symbol	Depiction of weld on drawings: in plan, face side	Depiction of weld on drawings: in plan, reverse side	Depiction of weld on drawings: in cross section
One-side single-V butt joint					
Two-side single-V butt joint					
Two-side double-bevel symmetric butt joint					
Two-side double-bevel asymmetric butt joint					
Two-side double-V symmetric butt joint					
Two-side double-V asymmetric butt joint					
Single-J joint					
One-side single-U butt joint					
Two-side double-U butt joint					

Lap Joints

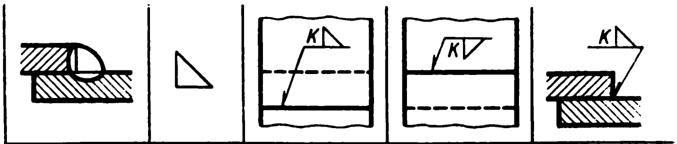
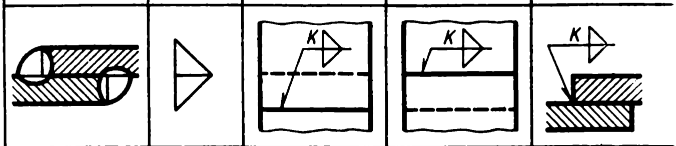
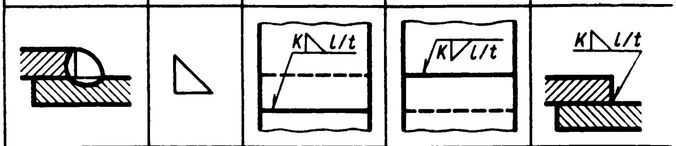
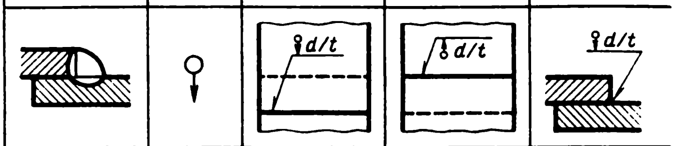
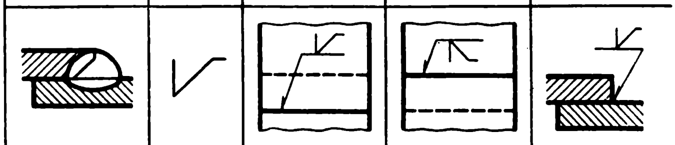
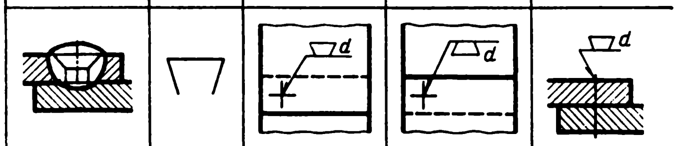
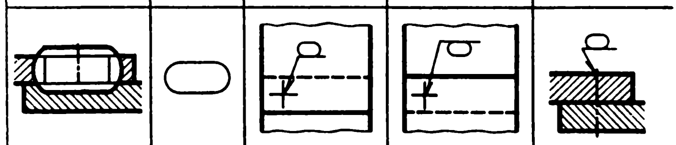
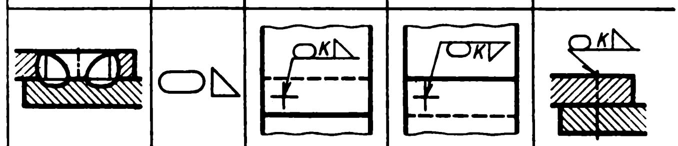
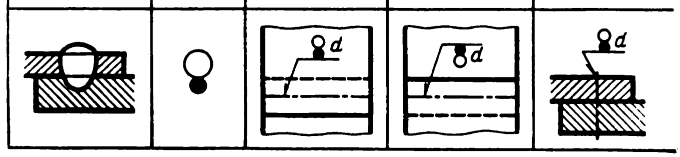
<table>
<tr><th rowspan="3">Type of joint</th><th rowspan="3">Weld</th><th rowspan="3">Symbol</th><th colspan="3">Depiction of weld on drawings</th></tr>
<tr><th colspan="2">in plan</th><th rowspan="2">in cross section</th></tr>
<tr><th>face side</th><th>reverse side</th></tr>
<tr><td>One-side square lap joint</td><td colspan="5">
</td></tr>
<tr><td>Two-side square lap joint</td><td colspan="5">
</td></tr>
<tr><td>One-side intermittent joint</td><td colspan="5">
</td></tr>
<tr><td>One-side spot lap joint</td><td colspan="5">
</td></tr>
<tr><td>One-side single-bevel lap joint</td><td colspan="5"></td></tr>
<tr><td>Circular slotted solid welded lap joint</td><td colspan="5">
</td></tr>
<tr><td>Linear slotted solid welded lap joint</td><td colspan="5"></td></tr>
<tr><td>Linear slotted incomplete lap joint</td><td colspan="5">
</td></tr>
<tr><td>One-side transfusion lap joint</td><td colspan="5">
</td></tr>
</table>

Table 8

Corner Joints

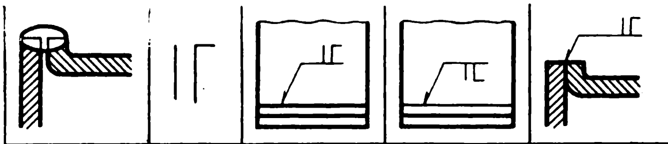
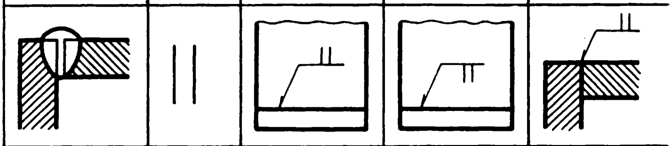
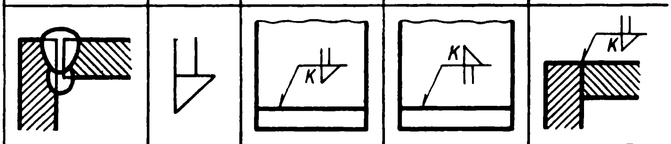
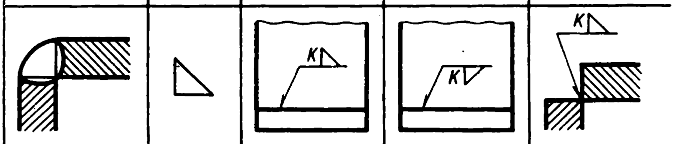
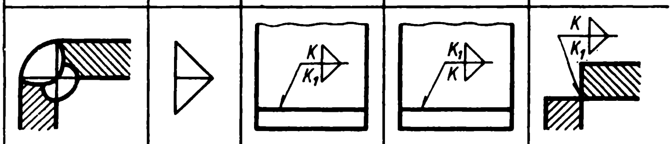
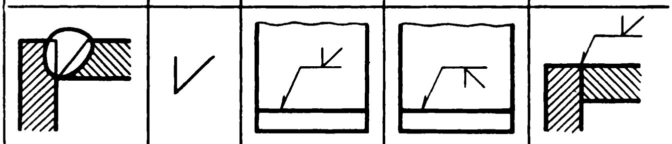
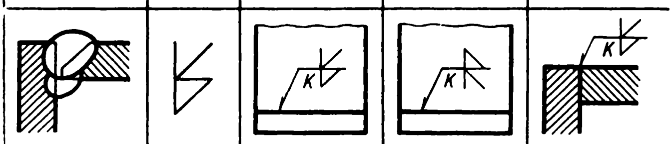
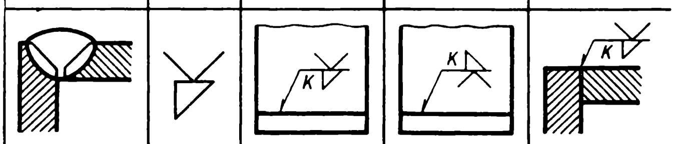
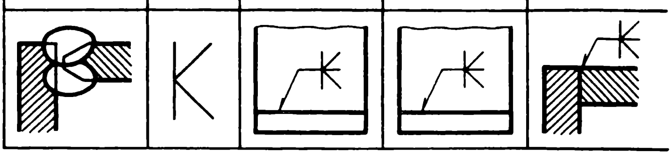
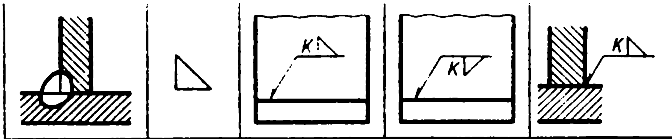
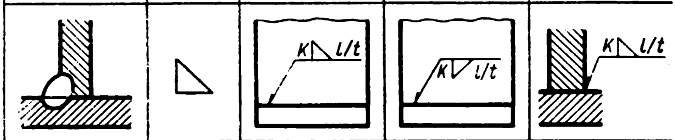
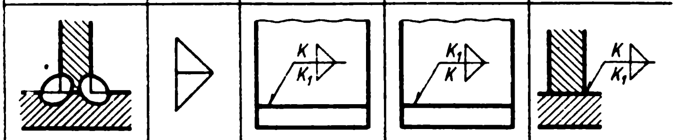
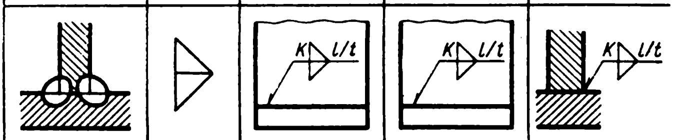
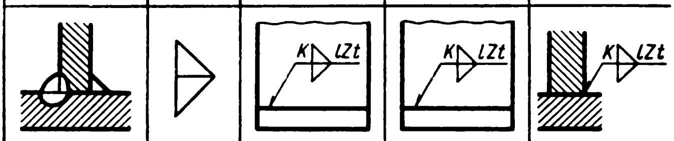
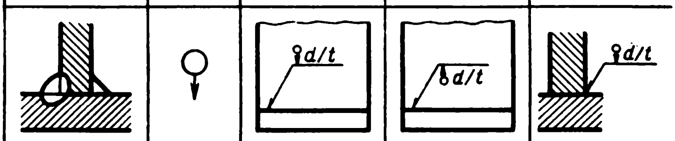
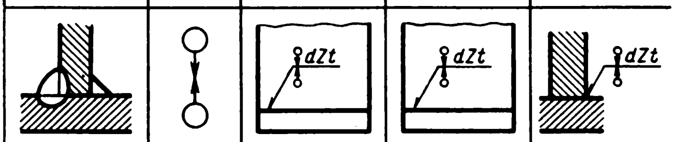
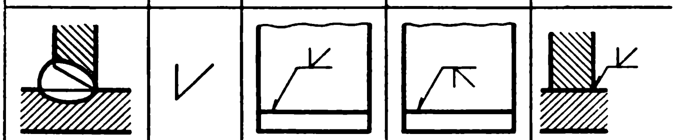
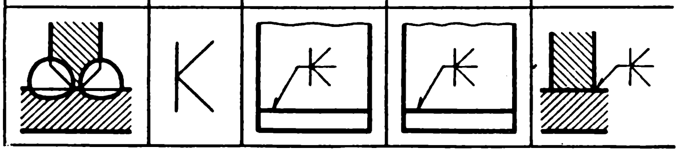
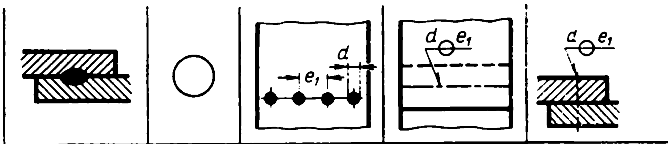
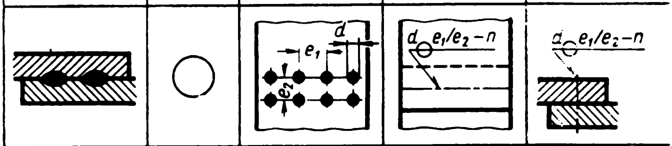
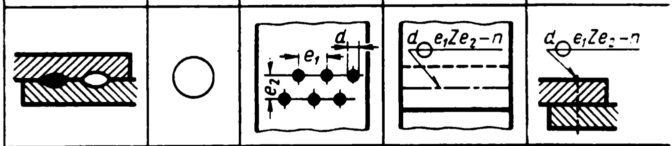
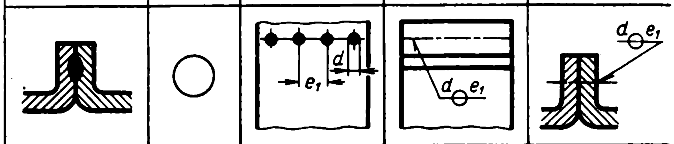
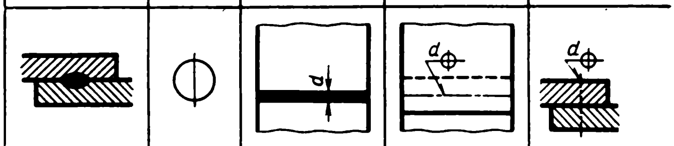
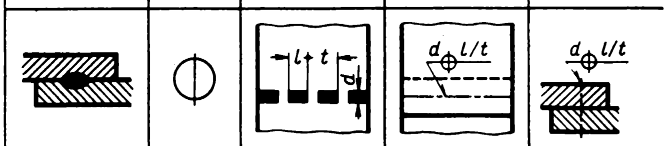
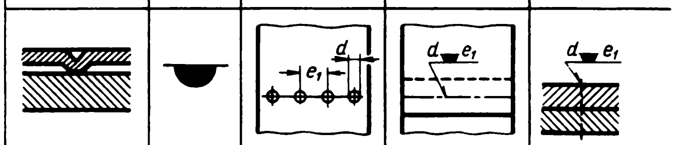
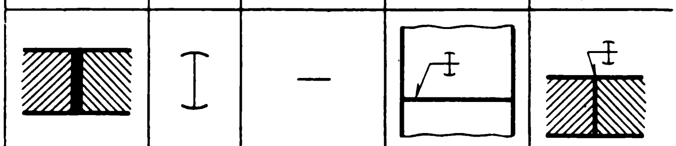
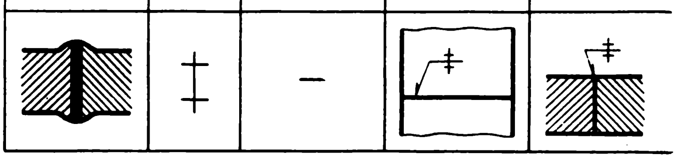
Type of joint	Weld	Symbol	Depiction of weld on drawings: in plan, face side	Depiction of weld on drawings: in plan, reverse side	Depiction of weld on drawings: in cross section
Flanged edge joint					
One-side square corner joint					
Two-side square corner joint					
One-side closed corner joint					
Two-side closed corner joint					
One-side single-bevel corner joint					
Two-side single bevel corner joint					
One-side single-V corner joint					
Two-side double-bevel corner joint					

Table 9

Tee Joints

Type of joint	Weld	Symbol	Depiction of weld on drawings: in plan, face side	in plan, reverse side	in cross section
One-side square tee joint					
One-side intermittent tee joint					
Two-side tee joint					
Two-side intermittent tee joint					
Two-side staggered tee joint					
One-side spot joint					
Two-side spot staggered tee joint					
One-side single-bevel tee joint					
Two-side double-bevel tee joint					

Joints Formed by Electric Resistance Welding

Type of joint	Weld	Symbol	Dimensioning diagram	Depiction of weld on drawings: in plan	Depiction of weld on drawings: in cross section
Single-spot joint					
Multiple-spot joint (*n* — number of rows)					
Staggered spot joint					
Double-flanged spot joint					
Seam joint					
Intermittent seam joint					
Projection welding					
Nonfusion butt welding			—		
Fusion butt welding			—		

This symbol is employed only for the simplest assembling units. In the case of intricate joints assembling drawings should be separately provided for each unit.

Welds intended for machining are marked by the finish symbol written on the extension line (Fig. 187*c*).

The symbol is used only in the simplest cases (cleaning of a weld). If machining changes the shape of the weld and affects the adjacent portions of the base metal a separate drawing (*weld assembly*) is provided which shows the product after welding with all necessary machining allowances, as well as the machining drawing (*mechanical assembly*) showing the product in its final form.

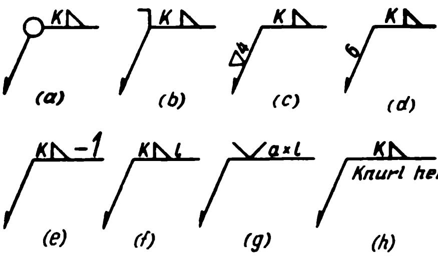


Fig. 187. Additional symbols

Welds of the same type and size are designated only once indicating the total number of welds of a given type (Fig. 187*d*), the other welds being marked only by extension lines.

If the welds are to be numbered according to the table on the drawing the ordinal number is written after the symbol (Fig. 187*e*). The figure should be 1.5-2 times higher than other symbols.

The length l of triangular fillet welds is marked as shown in Fig. 187*f*. The design thickness a of the sheets to be welded is also marked for other fillet welds (Fig. 187*g*). Additional data (for example, for strengthening processes) are written under the wing (Fig. 187*h*) or indicated by symbols which should be interpreted on the drawing or in technical documents.

Technical specifications use special denominations consisting of a letter indicating the mode of a welded joint (C, H, У, T—denoting butt, lap, corner and tee joints, respectively) and a figure specifying the type of a weld according to the USSR State Standard (ГОСТ 8713-58).

The method of designating the welds by a basic line is inconvenient for welds formed on separate portions of edges since the line of weld merges with the line of the contour and it is impossible to determine the length l of the weld (Fig. 188*a*) and coordinate the weld from the datum surface (size s) without additional explanations.

A weld can be shown by straight or slightly curved dash lines (Fig. 188*b*) with the height approximately equal to the width of the weld (to the scale of the drawing). The necessary dimensions are marked on the projection. Invisible welds are depicted by spaced lines.

Another method is to show the contours of a weld by thin lines, solid for visible welds and dash lines for invisible ones (Fig. 188*c*).

The drafting process is retarded if the welds are marked by bold lines (Fig. 188*d*) because it takes much time for the lines to dry. This method may be used only when each drawing is prepared individually, and also when welding drawings are rarely required for production.

The methods shown in Fig. 188*b-d* make it possible to indicate the dimensions of intermittent chain and staggered welds directly on the drawing (Fig. 188*e*) and also specify the distance *s* of the weld from the datum surface which is not shown by the symbols.

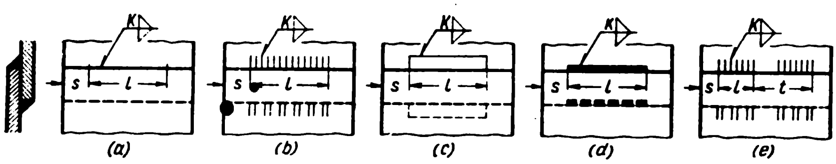


Fig. 188. Depiction of partial welds

The symbols of welds for which the edges are prepared must be explained in the technical specifications for welded joints and references to respective standards. For nonstandard welds, drawings should be prepared indicating the dimensions of all weld elements and edges (angle of edge preparation, clearance between edges, amount of edge blunting, height of reinforcement, etc.).

The drawings of lap joints should specify the width of lap, the distance of welds from longitudinal and transverse edges, and the dimensions and coordination of holes for plug joints.

5.3. Drawings of Welded Joints

The documents for welded joints usually include drawings of *blanks*, an assembly drawing of a welded joint (*weld assembly*), a machining drawing (*mechanical assembly*) and a drawing of the *welded part* in its final form.

An example of complete drawings of a welded structure is illustrated in Fig. 189.

Blanks (Fig. 189*a*, *b*) are drawn in the form in which they are delivered for welding with all the necessary allowances for subsequent machining of the joint.

Machined surfaces of blanks untouched by the machining of the welded joint are drawn in their final form indicating the needed tolerances and finish symbols.

On the weld assembly drawing (Fig. 189*c*) the product is shown as it should be after welding. Only parameters that are necessary for welding are specified: dimensions, type, length of welds, the dimensions showing the mutual arrangement of parts (without locating datum surfaces), and also the dimensions required to make welding jigs.

Superfluous dimensions (repeating the dimensions of the blanks, the dimensions being self-evident after connecting the parts by the locating datum surfaces) will only complicate the drawing and divert the attention of the worker.

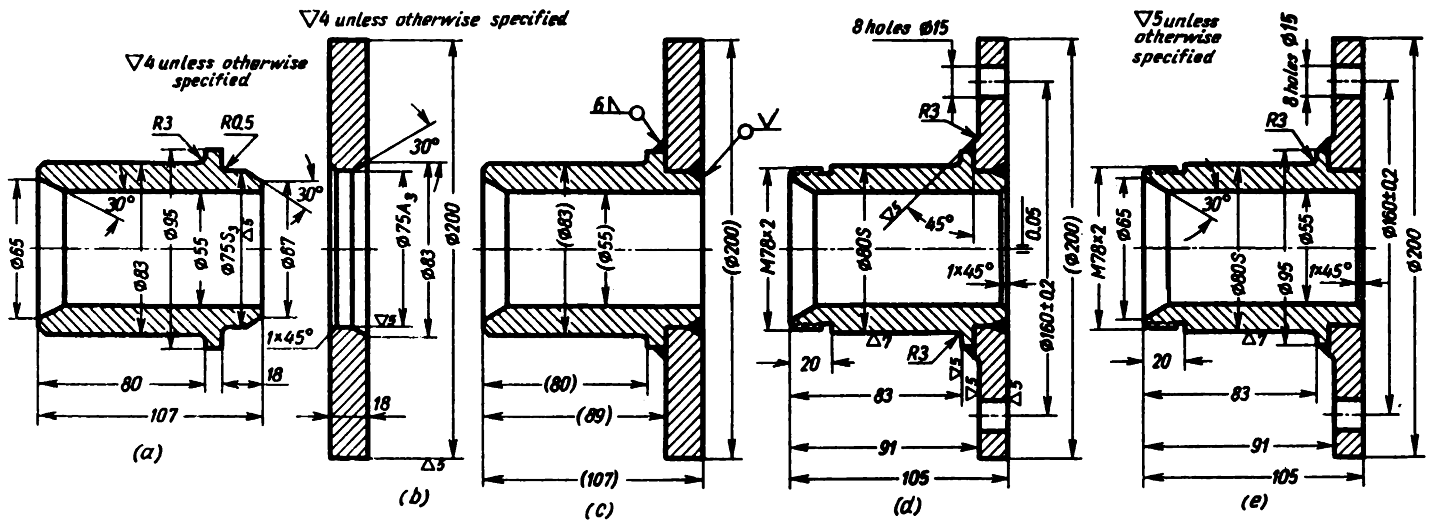


Fig. 189. Drawings of welded joints

a, *b*—blanks; *c*—weld assembly; *d*—mechanical assembly; *e*—welded part

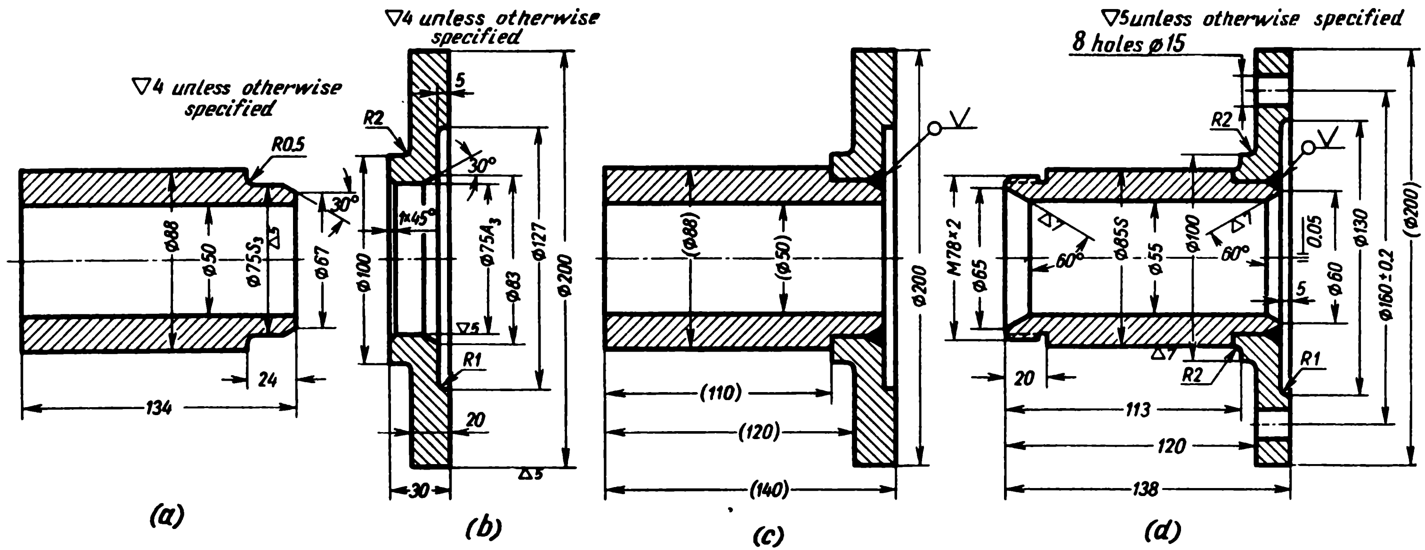


Fig. 190. Simplification of drawings of welded joints
a, *b*—blanks; *c*—weld-mechanical assembly; *d*—welded part

Reference dimensions are given in brackets to tell them from the dimensions to be strictly adhered to.

The overall dimensions of a unit should always be bracketed if they are intended for reference. The brackets are dispensed with if the dimensions should be maintained in the process of welding.

On a mechanical assembly drawing (Fig. 189*d*) the product is shown in the form it must have after machining, and all the work dimensions with the required tolerances should be marked on it. Other dimensions serve for reference.

The drawing of a welded part (Fig. 189*e*) should contain all the data necessary and sufficient for its application. Intermediate dimensions for welding and machining of blanks are omitted.

Figure 190 illustrates simplified methods.

If a welded product is machined by a circular (or almost circular) method (Fig. 190*a*-*c*) the mechanical assembly (Fig. 190*c*) may serve as the drawing for the welded part.

For simple welded joints made from shaped blanks (pipes, sheets, profiled rolled stock) it is usual to supply only an assembly welding drawing (Fig. 190*d*) on which all dimensions necessary for welding and the manufacture of blanks, as well as all data describing the product as a whole are marked.

When several subunits previously prepared are connected in one assembly it is expedient to make an assembly drawing of the joint specifying the data needed only for assembly.

The drawings of welded joints should indicate the total length of the welds of each type (as the basis for calculating the electrode consumption for the manufacture of the product).

The need for special tests of welded joints (for example, tests for air-tightness) is stipulated in the technical requirements of the drawing which also describes testing conditions, grounds for rejection and the methods of correcting faults.

5.4. Design Rules

Table 11 illustrates the rules for designing welded joints and shows examples of changes in designs with a view to improving manufacture of welded units.

5.5. Increasing the Strength of Welded Joints

The strength of welded joints can be increased by design methods (rational arrangement of welds with respect to the acting forces, proper form of welds) and by manufacturing methods (protection of the weld against harmful effects during welding, heat treatment, strengthening processing by cold plastic deformation). The design methods of increasing the strength are illustrated in Fig. 191.

Table 11

Rules for Designing Welded Joints

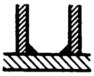
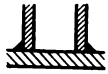
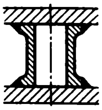
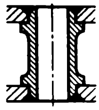
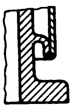
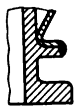
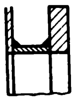
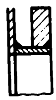
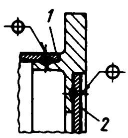
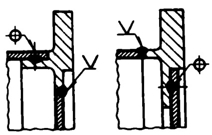
Design	
poor	improved
Ensure a convenient approach of electrodes to the weld	
Welding of partitions	
	Welds are brought out of the narrow space between the partitions 
Welding distance pipes to plates	
	Welds are brought out onto the surface of the plates 
Welding a jacket to a cylinder	
	Weld is brought away from the cylinder flange 
Welding a flange to a sleeve	
	Flange is removed from adjacent wall  Weld is brought to outer flange face 
Weld assembly of shell 1 with diaphragm 2	
 After one weld is completed it is difficult to seam weld the other	 One of the welds is made by the electric arc method

Table 11 (continued)

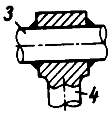
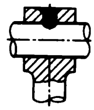
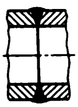
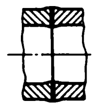
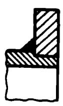
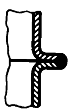
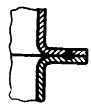
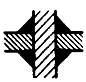
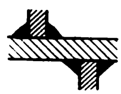
Design	
poor	improved
Employ the simplest and most efficient welding methods	
Connecting wrench handle 3 with bar 4	
	Circular welds are replaced by a rivet weld 
Connecting tubular parts	
	Electric arc welding by a circular weld is replaced by resistance butt welding 
C nnecting a flange to a pipe	
	Electric arc welding is replaced by resistance butt welding 
Welding of a tank	
	Electric arc welding is replaced by seam welding 
Avoid matched welds. Reduce the amount of built-up metal to the minimum	
Welding of ribs	
	Ribs are arranged in a staggered order 

Table 11 (continued)

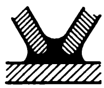
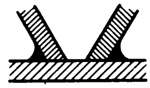
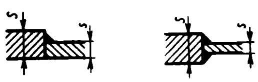
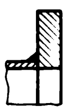
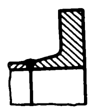
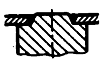
Design	
poor	improved
Welding of inclined partitions	
	Partitions are brought apart 
Avoid welding of thick parts with thin ones. Impart about the same cross sections to the edges being welded	
Limit ratios in butt welding	
 $S/s < 3$	 When $S/s > 3$ tapered portions of length $l > 5\ (S - s)$; $l' > 3\ (S - s)$ are introduced
Welding a flange to a thin-walled pipe	
	The flange is given a thin-walled annular transition portion 
Welding a pin to a plate	
	 The pin is given a thin-walled flange
	 Cutout is provided in the pin in the welding area

Table 11 (continued)

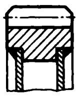
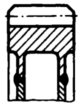
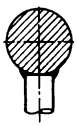
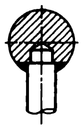
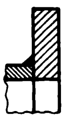
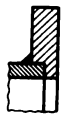
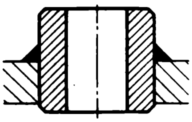
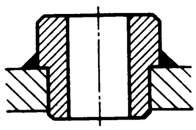
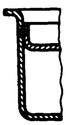
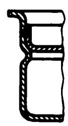
Design	
poor	improved
Welding disks to a gear rim	
	Rim is given thin-walled transition rings 
Arrange simple fixing of parts so that welding jigs are dispensed with	
	Head is centred on the bar 
Welding a flange to a pipe	
	Flange is centred on the pipe and held in the axial direction 
Welding a boss to a plate	
	Boss is fixed axially by shoulders 
Seam welding a partition to a shell	
	Partition is held in the axial direction by a flute 

Table 11 (continued)

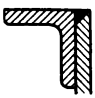
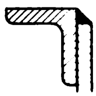
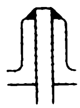
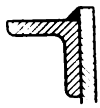
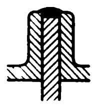
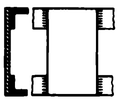
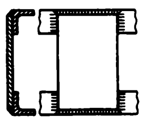
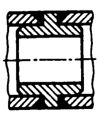
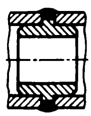
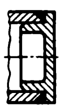
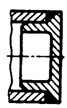
Design	
poor	improved
Avoid laborious edge preparation. Form welding pools by part displacement	
Welding of edges	
	
Corner joint	
	
Connecting shaped parts to plates	
 	   
Welding a corner plate	
	
Welding pipes to a coupling	
	
Process the parts which are simple to machine	
	Plug is machined 

Table 11 (continued)

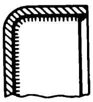
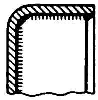
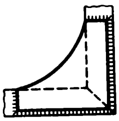
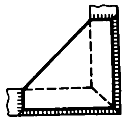
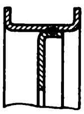
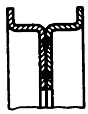
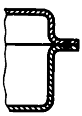
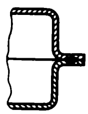
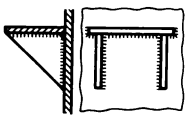
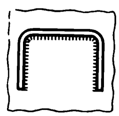
Design	
poor	improved
Eliminate fitting of preformed parts to complete joint contours. Simplify the preformed parts	
Welding a preformed rib to a trough-shaped profile	
	Rib is cut out at the fillet 
Gusset plate	
	The curved cut in the gusset plate is replaced by a straight one 
Unify the blanks	
Welded sheave	
	Sheave is made of two identical parts 
Tank	
	Tank halves are identical 
For thin-walled materials make wide use of bent and die-forged elements to increase the rigidity	
Welding of a flange	
	The composite flange is replaced by a formed one 

Table 11 (continued)

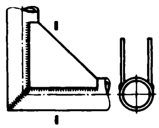
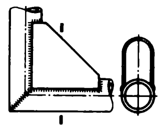
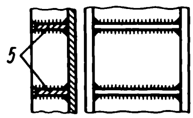
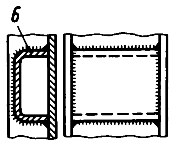
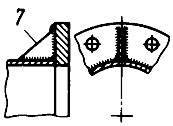
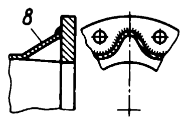
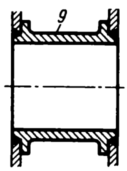
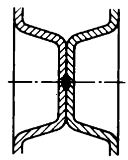
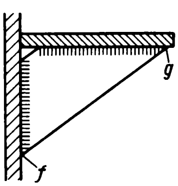
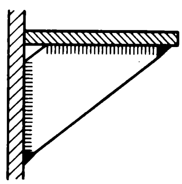
Design	
poor	improved
Reinforcing pipe corner joints	
	Separate flat gusset plates are replaced by one bent plate 
Reinforcing a trough-shaped profile	
	Welded-on ribs *5* are replaced by box *6* 
Connecting a flange to a pipe	
	Reinforcing ribs *7* are replaced by formed elements *8* 
Connecting of sheets	
	Spacers *9* are replaced by profiled elements 
Prevent burn and fusion of thin edges in the welding zone	
Welding of a rib	
	Sharp corners *q* and *f* are removed 

Table 11 (continued)

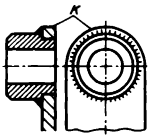
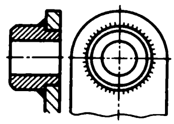
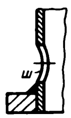
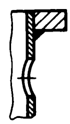
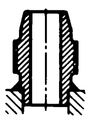
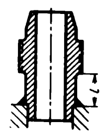
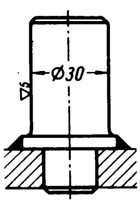
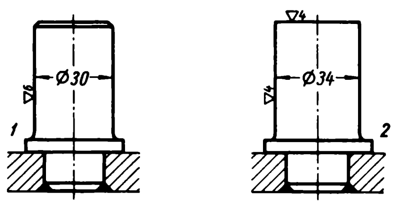
Design	
poor	improved
Welding a bushing to a lever	
	Burn of thin edge *k* is prevented by increasing its cross section 
Welding a flange to a ferrule	
	Fusion of the edge of hole *w* is prevented by removing the weld away from the hole. Another method is to drill the hole after welding 
Remove machined surfaces from the welding zone. Machine accurate surface after welding	
Welding of a threaded fitting	
	Thread is removed from the weld to a distance *l* sufficient to prevent fusion of the thread 
Welding of a pin	
	 1. Weld is removed from the machined surface 2. Stock on the pin is removed after welding

Table 11 (continued)

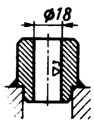
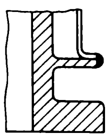
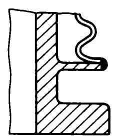
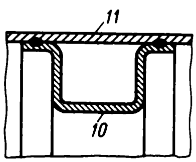
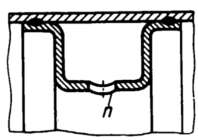
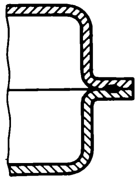
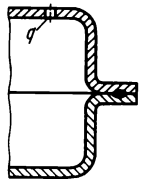
Design	
poor	**improved**
Welding of a bushing	
	 1. To prevent warping of the hole the weld is moved away from the body of the bushing 2. Hole is finish machined after welding
When parts with different cross section are welded, use heat buffers to prevent thermal stresses caused by nonuniform cooling	
Welding a jacket to a cylinder	
	The jacket is given elasticity by means of a crimp 
When welding closed cavities, prevent warping of walls caused by the formation of vacuum during cooling	
Welding an annular rigid profile 10 to shell 11	
	A ventilation hole n is provided in the profile 
Welded float	
	Hole q in the float is covered by weld after the flat has cooled 

Table 11 (continued)

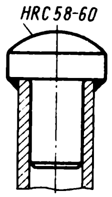
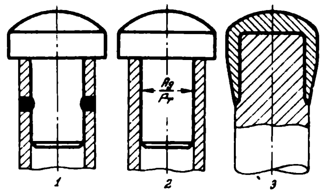
Design	
poor	improved
Do not weld together hardened and chemically heat treated parts (the effect of heat treatment is lost in heating)	
Connecting a hardened tip to a tubular rod	
	 1. Tip is connected by rivet welds 2. Welding is replaced by press-fitting 3. Head is stellitized

Figure 191, *1-3* shows consecutive strengthening of a torsionally loaded unit with a welded flange by increasing the diameter of the circular weld. The resistance to shear (proportional to the square of the joint diameter) with the same weld cross section is seven times larger in the design *2* and eighteen times larger in the design *3* than in the design *1*.

If the design of the weld is correct the additional fasteners (the thread, Fig. 191, *4*, the heavy drive fit, Fig. 191, *5*, etc.) may be dispensed with.

In centring joints the parts being welded are located by clearance fits usually with a class of accuracy not above the 3rd one (fits Se_3, Se_4, R_3, R_4, Rl_3). If more accurate centring is required use is made of slide fits S_{2a}, S_3 and wringing fits W_{2a}, W_3.

Welds should be relieved by transferring the load to sections with solid material, the welds being intended only to·join the parts.

Some examples of relieving the welds of loads are shown in Fig. 191, *6*, *7* (a bar loaded with an axial force) and in Fig. 191, *8*, *9* (bearing flange).

In the unit fastening the cover to the shell of a cylindrical reservoir subjected to internal pressure (Fig. 191, *10*) the welds of the cover and the shell are bent and shorn off by the pressure forces. In the improved design *11* the weld of the shell is relieved of internal pressure by introducing the shell into the flange and the weld of the bottom is relieved by clamping the bottom between the flanges of the shell and the bottom.

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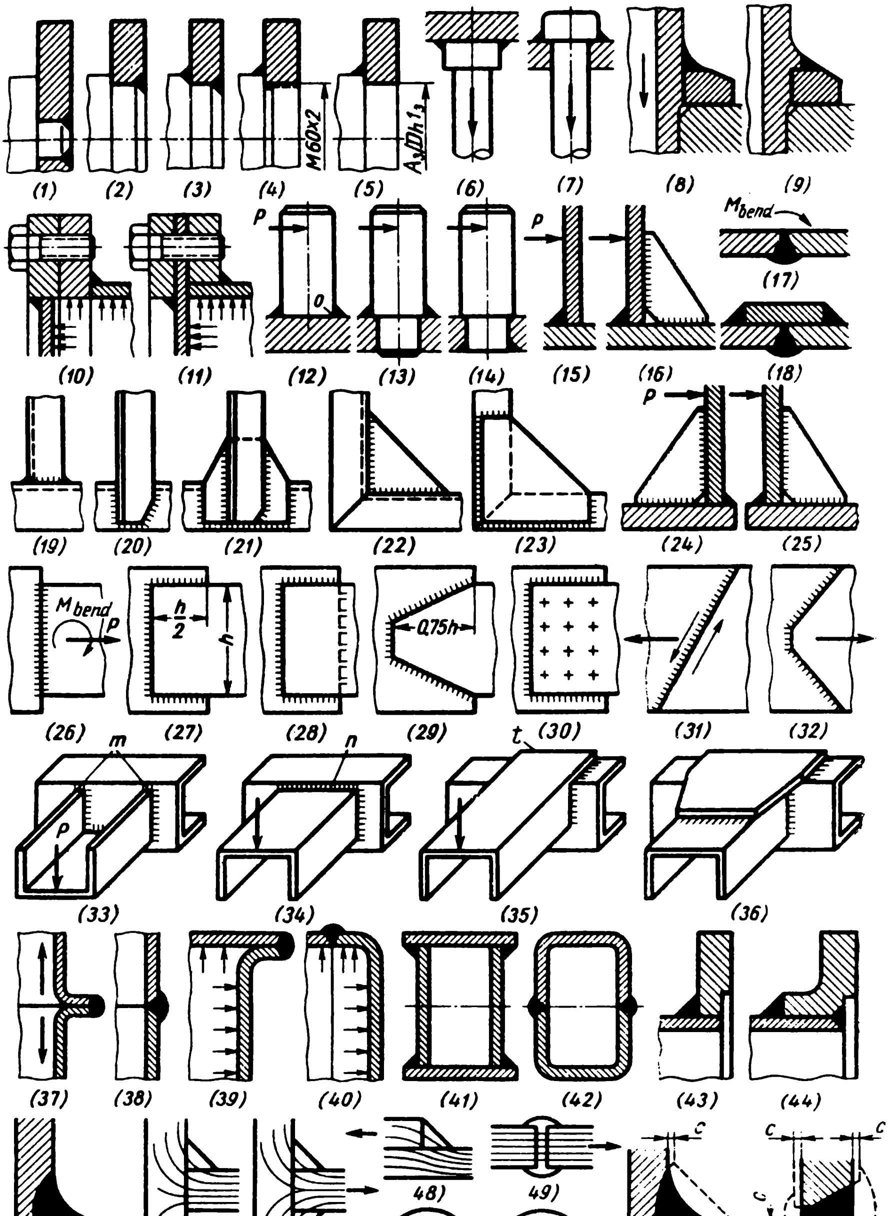


Fig. 191. Strengthening of welded joints

Power welds should preferably be loaded by shearing and tensile forces whereas bending load should be eliminated.

Figure 191, *12* shows an irrationally welded-on bar loaded with transverse force P. The force P rotates the bar about point O and produces high tearing stresses in the area opposite this point. Besides, the weld is subjected to shear.

Figure 191, *13* shows a better setup. The bar is centred in a seat of the part, and the weld is not subjected to shear. But the critical cross section of the bar is weakened by the weld.

In Fig. 191, *14* bending and shear caused by force P are acting upon the solid cross sections of the bar which are not weakened by welding. The weld is virtually relieved of the stress and is used only to secure the bar in the part.

It is better to reinforce with a rib the welded-on wall which is subjected to bending by force P (Fig. 191, *15*, *16*).

The bending of the butt-weld (Fig. 191, *17*) can be eliminated by using a strap (Fig. 191, *18*) whose welds are mainly in tension. In this design the butt weld is in compression.

The butt-weld of the angle bars (Fig. 191, *19*) is not strong enough. It is more reasonable to weld them over the plane of the flanges (Fig. 191, *20*) and strengthen them by corner plates for arduous operation conditions (Fig. 191, *21*).

It is better to join the corner plates not by butt welding (Fig. 191, *22*) but by lap welding (Fig. 191, *23*).

Welded-on ribs should be positioned so that they work in compression (Fig. 191, *25*) and not in tension (Fig. 191, *24*). This practically relieves the welds of all load.

Figure 191, *26-29* presents a consecutive strengthening of a sheet joint loaded by tensile force P and bending moment M_{bend}. The strength of various joints is compared in Table 12.

Table 12

Joint	Strength	
	tensile	bending
Butt joint (Fig. 191, *26*)	1	1
Lap joint (Fig. 191, *27*)	2	4
Lap joint with welded-on reverse side (Fig. 191, *28*)	3	5
Single-V lap joint (Fig. 191, *29*)	2.5	5

The strength of the butt joint shown in Fig. 191, *26* is assumed as a unit.

Besides all-round welding over the contour of long and thin plates, straps, corner plates, etc., should preferably be connected with

the basic member by further spot welding (Fig. 191, *30*) so that the plates may not come off when the system is deformed.

Skew welds of a lap joint (Fig. 191, *31*) subjected to tensile stresses are also affected by additional stresses emanating from shear along the line of the weld. In the balanced single-V joint (Fig. 191, *32*) the welds are relieved of shear.

Figure 191, *33-36* illustrates the weld designs of channel bar assemblies. In the joint with the channel legs arranged upwards (Fig. 191, *33*) the sections *m* of the vertical welds are subjected to high tearing stresses resulting from the action of force *P*.

When the channel bar has its legs downwards (Fig. 191, *34*) the load is taken by the long horizontal weld *n* and the weak end sections of the vertical welds are subjected to compression.

When the channel bar is connected by a tongue (Fig. 191, *35*) the welds are relieved of bending stress caused by force *P*. The bending moment is taken by longitudinal welds and the transverse weld *t* is in shear. Fig. 191, *36* shows a joint strengthened by a corner plate.

Out-of-centre force application causing a weld to bend should be avoided.

Flanged welds in units subjected to tension (Fig. 191, *37*) are bent. Butt-weld designs are better (Fig. 191, *38*). In the unit where a bottom is welded to a cylindrical reservoir with a flange (Fig. 191, *39*) internal pressure bends the weld. The butt weld (Fig. 191, *40*) is mainly subjected to rupture.

Welds should not be arranged in highly stressed zones.

In the case of welded beams subjected to bending it is good practice to arrange the welds not at the flanges (Fig. 191, *41*) but at the neutral line of the cross section (Fig. 191, *42*) where the normal stresses are the lowest.

In joints subjected to cyclic and dynamic loads, welds should not be made in sections where stresses are concentrated, for example in the transitions from one section to another (Fig. 191, *43*). In these conditions the weld is highly stressed and is also the source of an increased stress concentration due to the heterogeneity of its structure.

An improved design is illustrated in Fig. 191, *44*.

If it is impossible to move the weld beyond the section of stress concentration, concave welds should be used (Fig. 191, *45*) with deep penetration being attained by welding with a short arc.

The profile of a weld should be, as far as possible, symmetrical to the load action. Two-side welds (Fig. 191, *47*) are very effective in tee joints subjected to tension (Fig. 191, *46*). Butt joints (Fig. 191, *49*) should be used in preference to lap joints (Fig. 191, *48*). It is expedient to prepare the edges in butt joints on both sides

(Fig. 191, *51*) since the force lines are distorted in joints with an asymmetric weld (Fig. 191, *50*), with sharp stress variations.

The cyclic strength of welds can appreciably be increased by machining which imparts a rational form to the weld and reduces stress concentration.

It is good practice to machine corner welds radially with a smooth transition into the surfaces of the parts being joined (Fig. 191, *52*). Butt welds are machined flush with the surface of the product, the weld metal being removed both on the side of the basic weld and on the opposite side (Fig. 191, *53*).

For a smooth connection between a weld and the walls of a product it is necessary in most cases to undercut the walls simultaneously with the machining of the weld (dash lines on Fig. 191, *52*, *53*) providing for this purpose allowance *c*.

Figure 192 illustrates the cyclic strength curves for a strengthened butt joint (lower curves) and after the reinforcements are removed by machining (upper curves). Thin lines show the cyclic strength of the joint without heat treatment, and thick lines—after stabilizing heat treatment (annealing at 670 °C). The diagram shows that the removal of the reinforcements increases the cyclic strength approximately twice and the heat treatment by 15-20 per cent.

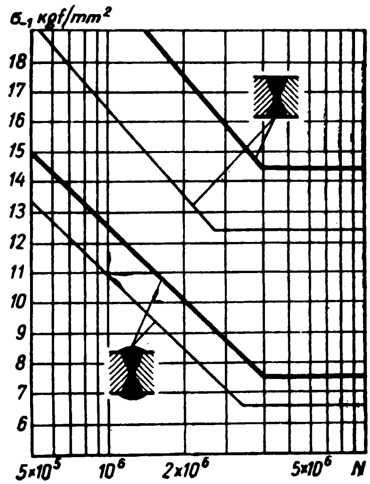


Fig. 192. Effect of heat treatment and machining of welds on cyclic strength. Steel 0X12НДЛ. According to Zaitsev G. Z. and Ponomarev V. Ya.

A smoothing fusion of welds with a tungsten electrode in argon medium considerably increases (by 30-40 per cent) the cyclic strength.

Plastic deformation in the cold state (roll burnishing, shot blasting, coining with pneumatic tools) makes it possible to raise the cyclic strength of the weld to the strength of the base metal.

5.6. Joints Formed by Resistance Welding

As a rule parts joined by butt resistance welding are not centred in relation to each other (Fig. 193*a*) because they are mutually fixed when mounted between the clamps of the welding machine and the upsetting mechanism. When the parts are centred (Fig. 193*b*) one of them should float in the clamps.

When thin parts are welded to thick ones transition sections corresponding to the form of the thin part being attached should be provided on the thick part (Fig. 193*c-e*, *f*, *g*).

If increased stability is to be ensured against bending the parts are joined in tapering seats (Fig. 193h). This design sharply reduces the force necessary to compress the parts when welding.

As distinct from electric arc welding, butt resistance welding makes it possible to join parts with machined surfaces (e.g., threaded

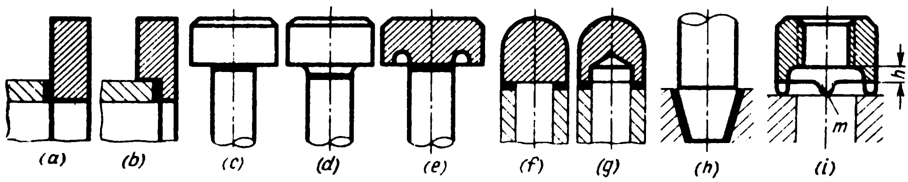


Fig. 193. Joints formed by resistance welding

members). The accurate surfaces should be located from the plane of the joint by distance of $h > 4\text{-}6$ mm (Fig. 193i) to prevent deformation and protect them against the sparks of molten metal. The amount of welded on metal and spark formation can be reduced and the consumption of electric energy decreased if welding is done with the use of separate projections m.

In the case of spot and seam welding of thin parts (less than 2 mm thick) the diameter of the spot and the width of the weld should

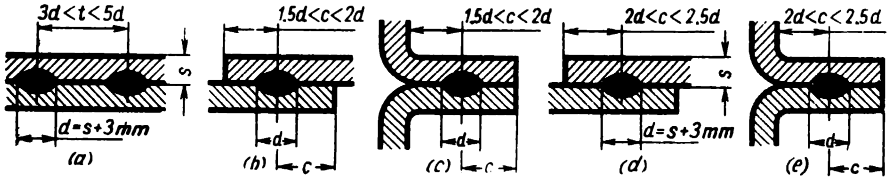


Fig. 194. Dimensions of spot and seam welds

be 2-3 times larger than the thickness s of the thinner element being welded. When thicker members are welded the diameter of the spot and the width of the weld are selected from the ratio $d = s + 3$ mm (Fig. 194a).

To avoid current shunting the pitch t of the spots should not be less than (3-3.5) d. The maximum pitch depends on the required strength and rigidity of the joint. The ratio $t < 5d$ should be maintained between the spots in order to prevent gaping of the plates.

The permissible distances c from the weld to the edges of parts being welded and to the adjacent walls are illustrated in Fig. 194b, c (spot welding) and d, e (seam welding).

The strength of spot and seam welds can appreciably be increased by compression of the spots and roll burnishing of seam welds under pressure that slightly exceeds the yield point of the material.

5.7. Welding of Pipes

Pipes of equal diameter are commonly joined with a butt fillet weld without edge preparation (Fig. 195, *1*), and with edge preparation when the walls are thick (Fig. 195, *2*).

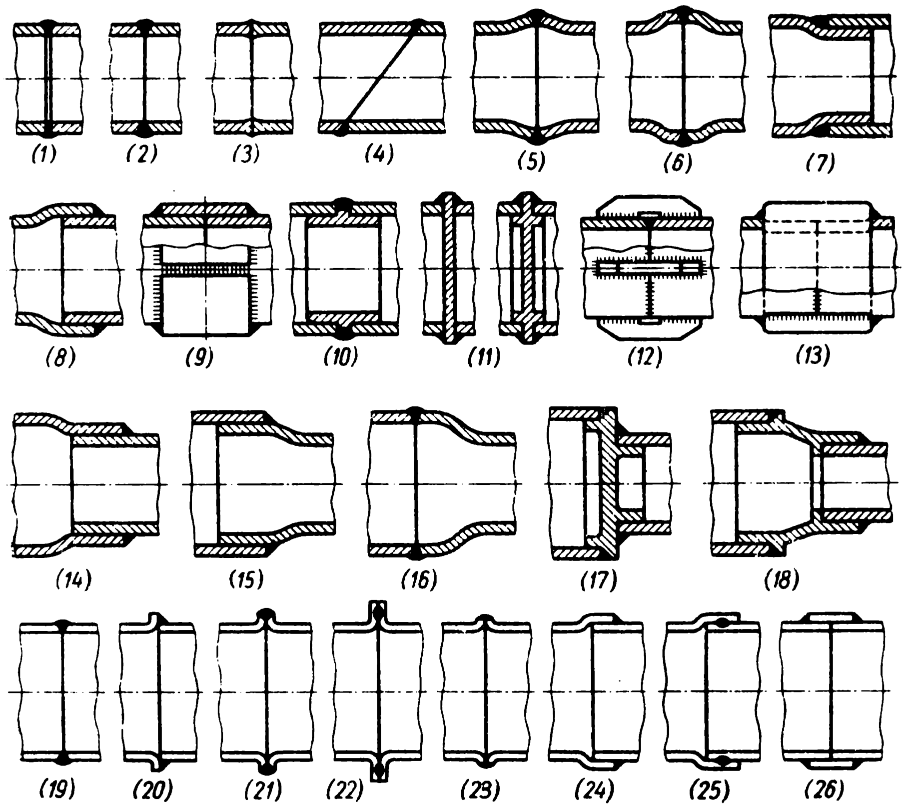


Fig. 195. Welding of pipes

A joint formed by butt resistance welding (Fig. 195, *3*) is distinguished by its high strength, but is difficult to make at the assembly site.

A skew joint (Fig. 195, *4*) is technically unsound, and does not increase joint strength.

The abutting ends of a pipe are expanded to a taper (Fig. 195, *5*) or to a bell (Fig. 195, *6*) to increase the bending strength.

This purpose is also served by compressing (Fig. 195, *7*) or expanding (Fig. 195, *8*) one of the pipes. The latter method is preferred since it is easier to expand a pipe than compress it.

Figure 195, *9* shows a joint reinforced with an external sleeve.
Internal sleeves (Fig. 195, *10*) reduce the active pipe diameter, thus making this method undesirable for pipelines; it is however used for force-loaded structures, where strong and rigid connection with diaphragms is necessary (Fig. 195, *11*).
Reinforcement of a joint by ribs (Fig. 195, *12*) spoils the external appearance and is also inferior in strength to other joints.
A joint with cut-in ribs (Fig. 195, *13*) is stronger, but more complicated in manufacture.
Figure 195, *14-16* shows methods of connecting various diameters pipes when the difference between them is small.
When the difference in the diameters is considerable intermediate inserts (Fig. 195, *17*) are introduced. Taper inserts (Fig. 195, *18*) are highly rigid and permit one to connect pipes with a large difference in the diameters.
Thin-walled pipes are butt-welded with a fillet weld (Fig. 195, *19*) accomplished preferably by gas welding with flanging of one (Fig. 195, *20*) or two (Fig. 195, *21*) edges, and also by seam welding (Fig. 195, *22*). If the diameter and the length of pipes are such as to admit electrodes, use is made of seam welding over the flanged edges (Fig. 195, *23*).
The joints are reinforced by the expansion (Fig. 195, *24*, *25*) or by sleeves (Fig. 195, *26*).
The joints in Fig. 195, *24-26* are centred. Other joints must be centred during welding.

5.8. Welding-on of Flanges

The methods of welding flanges to pipes are illustrated in Fig. 196.
The shortcoming of the design *1* in Fig. 196 is that the flange is not fixed radially.
In the designs *2* and *3* the flange is not secured axially. When mounted on rough pipe surface (and therefore with a large clearance) the flange may misalign during welding. Besides, in such designs, the weld comes out onto the flange face and is partly cut off when the flange is machined.
In the design *4* the flange is locked in the radial and axial directions by a machined step and is insured against misalignment by being located against the end-face of the step.
Figure 196, *5-7* shows joints where the weld is not on the flange end-face.
Electric resistance welding (Fig. 196, *8*, *9*) is the most simple and productive method.
The methods of welding flanges to thin-walled pipes are presented in Fig. 196, *10-14*. Design *11* is superior to design *10* as the flange is secured both in the radial and axial directions.

Seam welding (Fig. 196, *12*) is employed when the diameter of the pipe allows a roller electrode to be introduced inside the pipe.

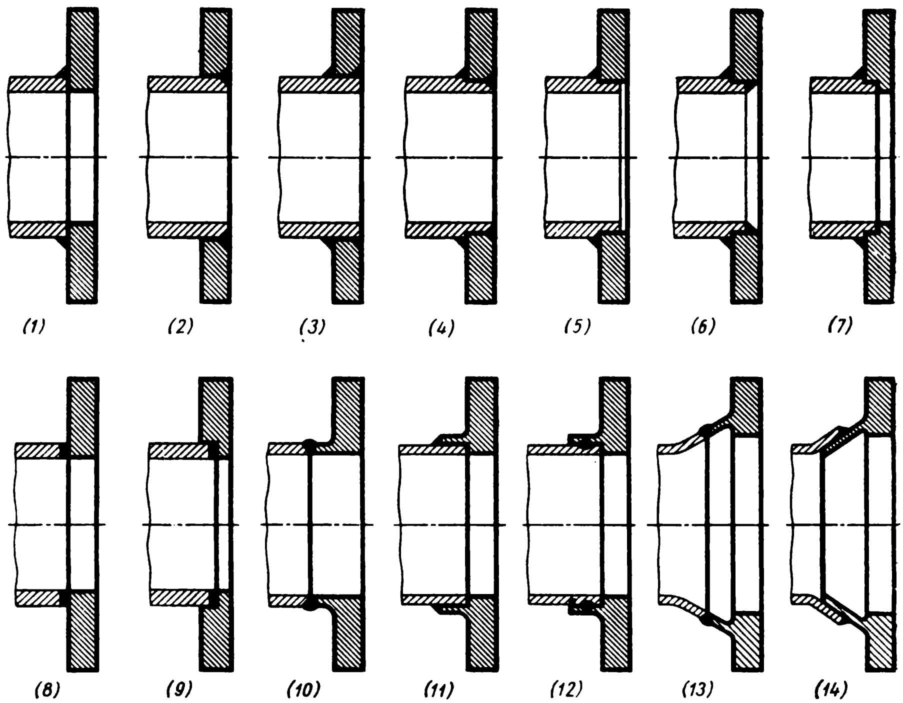


Fig. 196. Welding-on of flanges

Figure 196, *13*, *14* illustrates the bell-mouth methods of welding usually used to join large-diameter flanges.

5.9. Welding-on of Bushings

Figure 197, *1-6* shows how threaded bushings can be connected to flat plates.

In the design *1* in Fig. 197 the bushing is not centred in relation to the plate. The internal threaded surface of the bushing is deformed during welding. This shortcoming is corrected in the design *2*. In the most rational design *3* the weld is removed from the body of the bushing.

Spot or seam welding (Fig. 197, *4*) is employed when the diameter of bushings is large.

Butt resistance welding (Fig. 197, *5*) is remarkable for its high productivity and does not damage the thread.

It is better to flange thin-walled sheets to suit the bushing contour (Fig. 197, *6*).

Figure 197, *7-18* shows some methods of welding bushings to the walls of cylindrical shells.

It is undesirable to weld the flat surface of a bushing to a cylindrical surface (Fig. 197, *7*), since the bushing is distorted during welding and the weld is vague and variable in thickness.

A better design is shown in Fig. 197, *8* where the end-face of the bushing is chamfered to obtain a more proper shape of the weld.

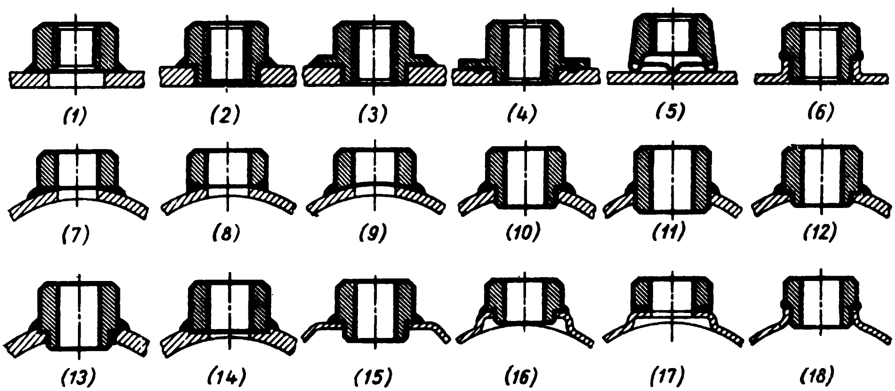


Fig. 197. Welding-on of bushings

The design in Fig. 197, *9*, in which the surface of a bushing is machined over a cylinder to a radius equal to that of the shell, is technically unsound and is of no use if the bushing has to be centred in the shell.

Figure 197, *10-14* illustrates some methods of welding with centring of the bushing.

In Fig. 197, *10* the weld varies in thickness.

In Fig. 197, *11* where the bushing is put through a hole in the shell, the bushing must be supported during welding or first clamped in position. Misalignment can occur during installation.

If the wall of the shell is thick enough a correct joint can be obtained by making a flat (Fig. 197, *12*) or facing the wall up (Fig. 197, *13*,*14*).

In the case of thin-walled shells a correct weld can be accomplished by local wall deforming (Fig. 197, *15-17*).

The most useful is the design *18* where the shell walls are flanged and the flange ends are then machined or cleaned.

Figure 198 presents methods of welding circular flanges to cylindrical shells.

In Fig. 198, *1* the surface of the flange being attached is machined to a cylinder. To prevent warping of threaded holes they are machined after welding (Fig. 198, *2*).

In Fig. 198, *3* the weld is separated from the body of the flange by a shoulder made integral with the flange. Such flanges are die-forged.

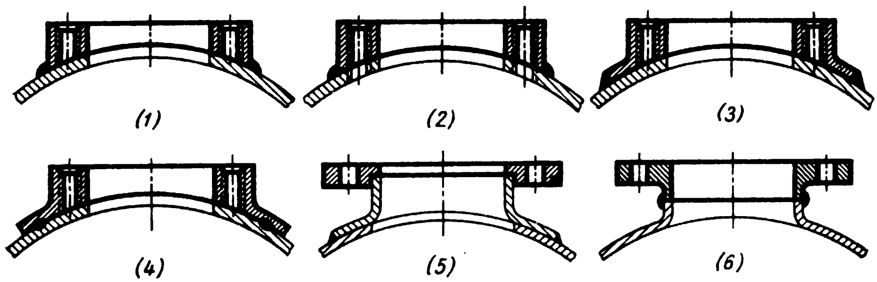


Fig. 198. Welding of flanges to a shell

It is difficult to weld-on a flange by spot electric welding (Fig. 198, *4*) because of the spatial arrangement of the weld. Seam welding is still more complicated.

Figure 198, *5*, *6* shows the ways flanges can be welded to thin-walled shells.

5.10. Welding-on of Bars

Bars are welded to massive parts and thin sheets usually by resistance welding. This method is frequently utilized to attach studs to steel parts and parts made of high-strength cast iron. In large-scale production welding is much more advantageous than the common method of fastening with threaded studs.

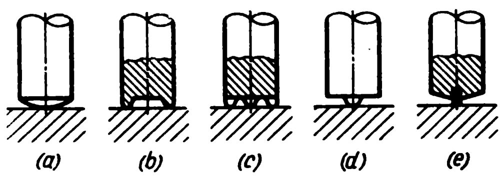


Fig. 199. Welding-on of bars

The consumption of electric energy will be reduced if welding is performed over a restricted perimeter or at individual spots. The ends of bars are made spherical (Fig. 199*a*), provided with annular rims (Fig. 199*b*) or projections (Fig. 199*c*, *d*).

Large-diameter bars (over 8 mm) are welded with the use of flux. In mass production solid flux inserts are first introduced into the bars (Fig. 199*e*).

Flash welding with the use of flux is employed to attach bars of diameters up to 25 mm. A ceramic bushing (Fig. 200*a*-*c*) placed on the bar retains the molten flux and metal and limits the contour of the weld.

The energized bar is brought to the welding position (Fig. 200*a*) and an electrical arc is struck after which the bar is drawn away to a distance of 0.5-1 mm (Fig. 200*b*) and kept in this position for the time sufficient to melt the metal of the bar and the part. Then, the bar is immersed into the molten metal pool (Fig. 200*c*) and the entire cross section of the bar is welded (Fig. 200*d*). The duration of the process is 0.1-1 second.

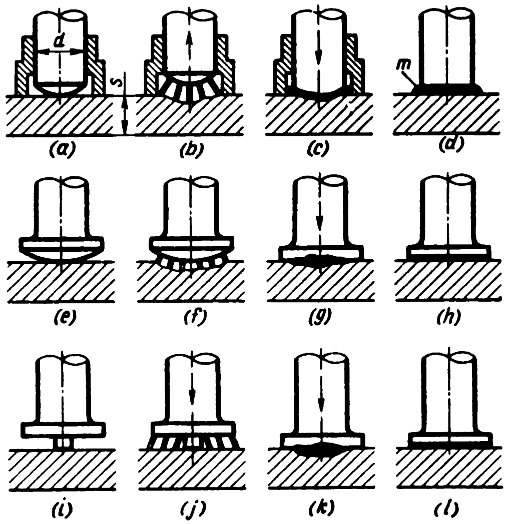


Fig. 200. Connection of bars by flash welding

The annular welded metal collar *m* formed on the periphery of the bar is overlapped when the parts are joined with the use of holes of increased diameter, by chamfering the edges of the hole or placing thick gaskets in the joint.

In the case of welding without a support the minimum permissible thickness of the plate $s_{min} \approx \approx 0.5d$ (where *d* is the bar diameter), and with a support — $s_{min} \approx 0.3d$.

To avoid shunting of the current the distance between the adjacent bars should be at least (3-3.5) *d*.

The method of *condenser welding with an impulsive discharge* does not require the use of flux and permits parts of heterogeneous materials to be joined.

The bar is pressed by a spring against the plate (Fig. 200*e*) and an electric impulse is supplied that melts the metal at the joint (Fig. 200*f*). The force of the spring presses the bar into the molten metal (Fig. 200*g*) and the joint without a weld collar is formed (Fig. 200*h*).

A variety of this process is welding with the use of melting stud (Fig. 200*i-l*).

Condenser welding can be used to attach bars of diameters up to 10 mm. The thickness of the plate and the distance between the bars are practically unlimited.

Milliseconds are required to complete the process. Automatic welding machines operate at a rate of 100 welding operations per minute.

5.11. Welded Frames

Figure 201, *1-18* illustrates the methods of welding frames made of angle bars.

Joints with angle bars arranged with their vertical flanges outwards (Fig. 201, *1-6*) are most popular. They ensure a smooth external form of the frame.

The most common design is a butt joint in which the edges are bevelled at an angle of 45° (Fig. 201, *1*). Tenon welds with cuts in the flanges of the angle bars are much more complex (Fig. 201, *2-4*).

Figure 201, *5* shows connection of edges in which the external corner of the joint is rounded. A strong joint can also be obtained when the corners are bent over a solid wall with the flanges cut and connected at an angle of 45° (Fig. 201, *6*).

The corners with the inward arrangement of their vertical flanges (Fig. 201, *7-12*) spoil the external appearance of the frame, but make it easier to fasten diagonal ties.

The most frequent designs are butt joints with flanges bevelled at an angle of 45° (Fig. 201, *7*) usually in combination with strengthening corner plates (Fig. 201, *8*).

Figure 201, *9-10* illustrates butt joints with straight edges. The joint in Fig. 201, *10* can be reinforced by a corner plate (Fig. 201, *11*) which cannot be used in the joint in Fig. 201, *9*.

Figure 201, *12* shows a tongue-welded edge joint.

The methods of connecting frames with a combined arrangement of angle bars (one bar with the flange inside and the other outside) are shown in Fig. 201, *13-18*.

The diagonal ties in frames with the inward arrangement of vertical flanges of the angle bars are butt-welded to the walls of the bars with the edges at an angle of 90° (Fig. 201, *19*). The joint can be strengthened by a corner plate (Fig. 201, *20*). Tubular ties are fastened in a similar manner (Fig. 201, *21*).

When the angle bars are arranged with their vertical flanges outwards the diagonal ties are fastened by means of corner plates (Fig. 201, *22*). The butt connection with a shaped cut of the edges (Fig. 201, *23*) is not technically sound and is less strong than a joint with corner plates.

Corner braces (Fig. 201, *24*) are frequently used instead of diagonal ties. Like the latter they can be welded on easier when the angle bars of a frame are positioned with the inward arrangement of their vertical flanges.

A crosswise connection of diagonal ties in the centre of a frame (Fig. 201, *25-30*) is difficult especially if the ties are made of asymmetric profiles (for example, angle bars).

A joint formed of solid angle bars welded on the flanges (Fig. 201, *25*) is simple and strong enough, but the shortcoming of the design is

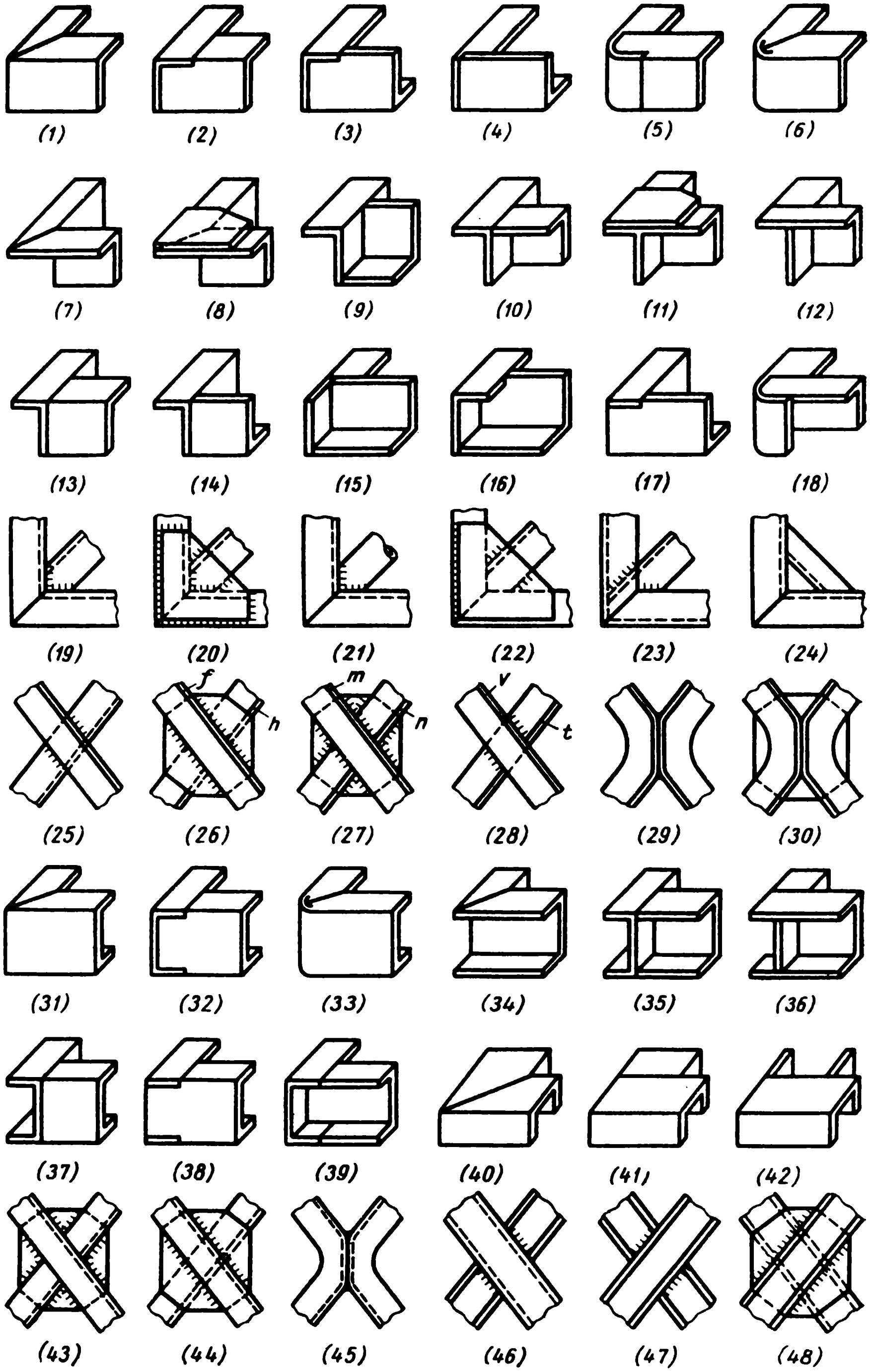


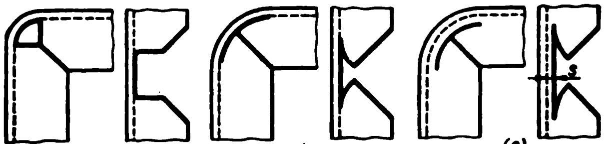
Fig. 201. Welding of profiled frames

that the height of the flange of diagonal angle bars should be half that of the basic angle bars of the frame.

In the design *26* in Fig. 201 the bar *f* is solid and bar *h* is cut. The flanges of the bars have their faces in opposite directions and are welded to the plate arranged between the flanges. The height of the angle bars in this design may be equal to the height of the basic bars of the frame minus the thickness of the gusset plate.

In the design *27* in Fig. 201 the solid angle bar *m* and the cut bar *n* face with their flanges in one and the same direction and are welded to each other with the use of the gusset plate. The diagonal bars may be identical to the basic bars of the frame, with the gusset plate protruding beyond the plane of the frame.

The rib of bar *t* in design *28* in Fig. 201 is cut out for the flange of bar *v*. The strength of the joint is inferior to that of the previous



(a) (b) (c)

Fig. 202. Methods of bending angle bars

two joints. The height of the angle bars may be equal to that of the basic bars of the frame minus the thickness of the flange.

In the design in Fig. 201, *29* bent angle bars are welded together by their flanges. In this case the diagonal angle bars may be identical to the basic bars of the frame. The joint can be reinforced by gusset plate (Fig. 201, *30*).

Figure 201, *31-33* illustrates the methods for connecting channel-bar frames with inward flanges, Fig. 201, *34-36*—with outward flanges, Fig. 201, *37-39*—with a mixed arrangement and Fig. 201, *40-42* — with flanges arranged perpendicular to the plane of the frame.

Some methods of crosswise connection of diagonal ties made of channel bars in a "standing" position are represented in Fig. 201, *43-45*, and in a "lying" position—in Fig. 201, *46-48*.

The methods of bending angle bars by cutting the flanges are shown in Fig. 202.

In the design in Fig. 202*a* with a right-angled cut a triangular hole is formed upon bending which may be welded in or closed with a corner plate.

Full closure of the edges is ensured by the shaped cutout illustrated in Fig. 202*b*.

In the design in Fig. 202*c* the cut is removed from the wall of the angle bar by distance *s* slightly exceeding the radius of the fillet between the internal walls of the bar. This makes cutting out easier and increases the strength of the joint.

Tubular frames are usually connected by butt joints with tube ends bevelled at an angle of 45° (Fig. 203, *1*).

The rigidity of the corners is increased by flattening the ends of the tubes (Fig. 203, *2*), butt welding of gusset plates (Fig. 203, *3*) or by cut-in welds (Fig. 203, *4*) using gusset double plates (Fig. 203, *5*), bent U-plates (Fig. 203, *6*), shaped plates (Fig. 203, *7*) consisting of two halves which enclose the tubes and are welded around the tubes being joined by spot welding.

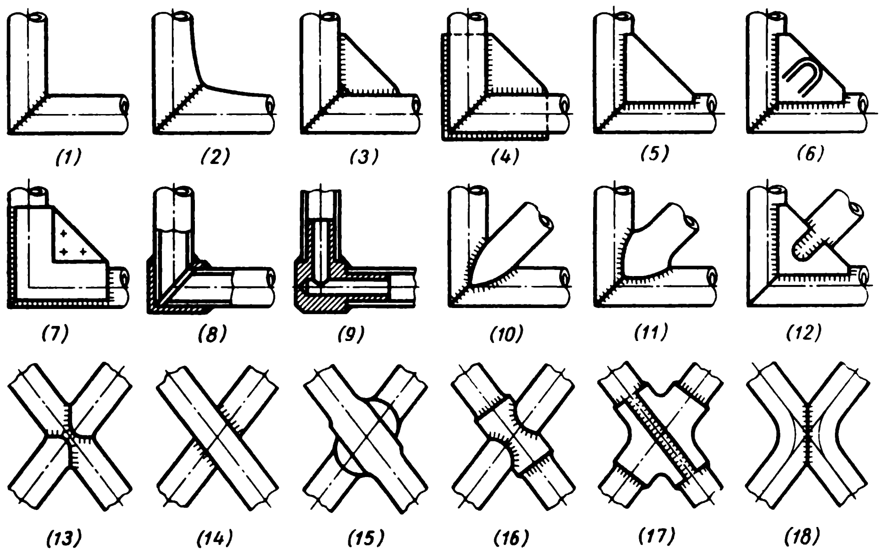


Fig. 203. Welding of tubular frames

Figure 203, *8* shows a strong but expensive joint with a die-forged angle bar with holes into which the 45° cut tube ends are introduced. In Fig. 203, *9* the angle bar has necks to which the tubes are welded.

Tubular diagonal ties are butt-welded to the corners of frames (Fig. 203, *10*), with flattening of the diagonal tube (Fig. 203, *11*) and strengthening the joint by a U-shaped slotted plate to which the diagonal tube is welded (Fig. 203, *12*).

Intersecting joints of diagonal tubular ties are butt- (Fig. 203, *13*) or cross-halving (Fig. 203, *14*) welded with a cut in one or both tubes. Other methods are: upsetting the tubes at the joint connection (Fig. 203, *15*), connection by means of a cylindrical sleeve (Fig. 203, *16*) and connection by formed sheet straps (Fig. 203, *17*). Figure 203, *18* shows connection of bent pipes when they are flattened at the joint. Another method is cutting the tubes at the joint plane and then welding.

5.12. Welded Truss Joints

In the units with angle bars (Fig. 204. *1*) butt-connections should be avoided. Lap joints (Fig. 204, *2*) with the contour of the angle bar welded all around are stronger and more rigid. Flanges of the bars are advisably crossed perpendicular to the plane of the joint. The designs in Fig. 204, *4*, *6* are much more rigid than the joints in Fig. 204, *3*, *5*.

To avoid excessive bending and torsional moments the truss elements should be connected so that the bending centre lines of all the sections in the unit intersect at one point (designs in Fig. 204, *7*, *9* are wrong and designs *8* and *10* are correct).

The bending centre lines should also align in its transverse plane. Joints with flanges facing one way (Fig. 204, *11*, *12*) are better than those with flanges facing in opposite directions (Fig. 204, *13*, *14*). In the latter case under load a twisting moment develops in the unit due to the displacement of the bending centre lines.

When the flanges face one way the design is more compact. In the designs in Fig. 204, *11*, *12* the width of the unit (in the plane normal to the plane of the drawing) is about two times less than in the designs in Fig. 204, *13*, *14*. However, the units and the truss as a whole in the designs in Fig. 204, *13*, *14* are more rigid spatially. The formation of the welds is simpler and this makes such designs very popular in practice.

The rigidity of a joint can be improved by gusset plates. A joint with strapped gusset plates (Fig. 204. *16*) is much stronger and more rigid than a joint with abutting gusset plates (Fig. 204, *15*).

Figure 204, *17-18* exemplifies multi-ray joints with strapped gusset plates. The comparative advantages and shortcomings of flanged joints facing one (Fig. 204, *17*) or two (Fig. 204, *18*) ways are the same as for joints without gusset plates (Fig. 204, *11-14*).

Figure 204, *19-22* shows examples of joining angle bars in spatial units.

Butt welding (Fig. 204, *23*, *24*) forms the simplest and most reliable joints in tubular trusses. The shortcoming of the method is the limited number of tubes that can be connected in one unit. Spatial units are possible only if the diameter of the central tube considerably exceeds the diameter of the attached tubes (Fig. 204, *25*).

If the tubes being joined are flattened (Fig. 204, *26*, *27*) it is possible to increase the number of the tubes connected in one unit (Fig. 204, *28*) and increase the rigidity of the joint (only in the flattening plane).

When tubes of different diameters are joined, the tube of the smaller diameter is conically expanded (Fig. 204, *29*, *30*) to increase the rigidity of the unit.

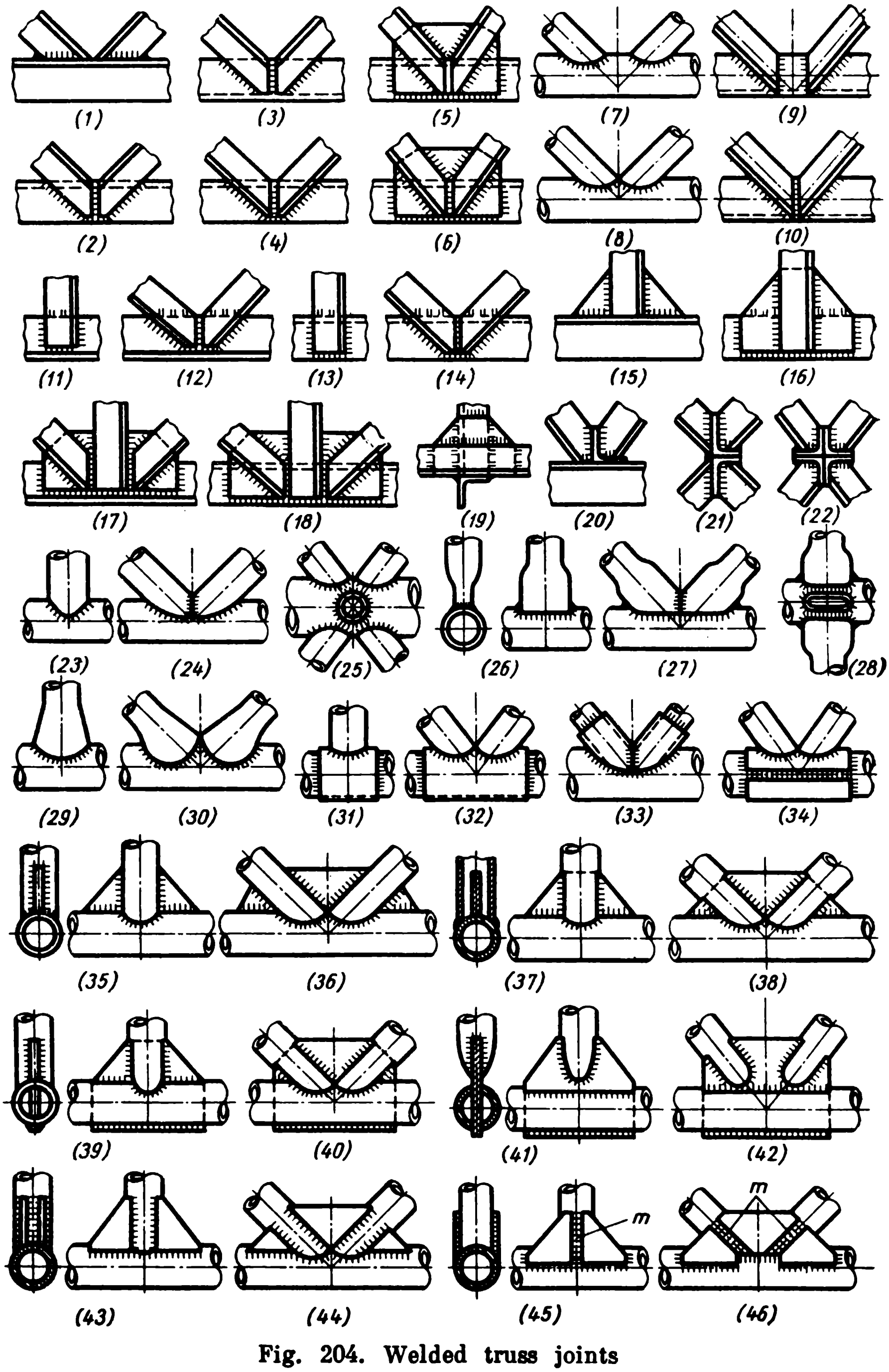


Fig. 204. Welded truss joints

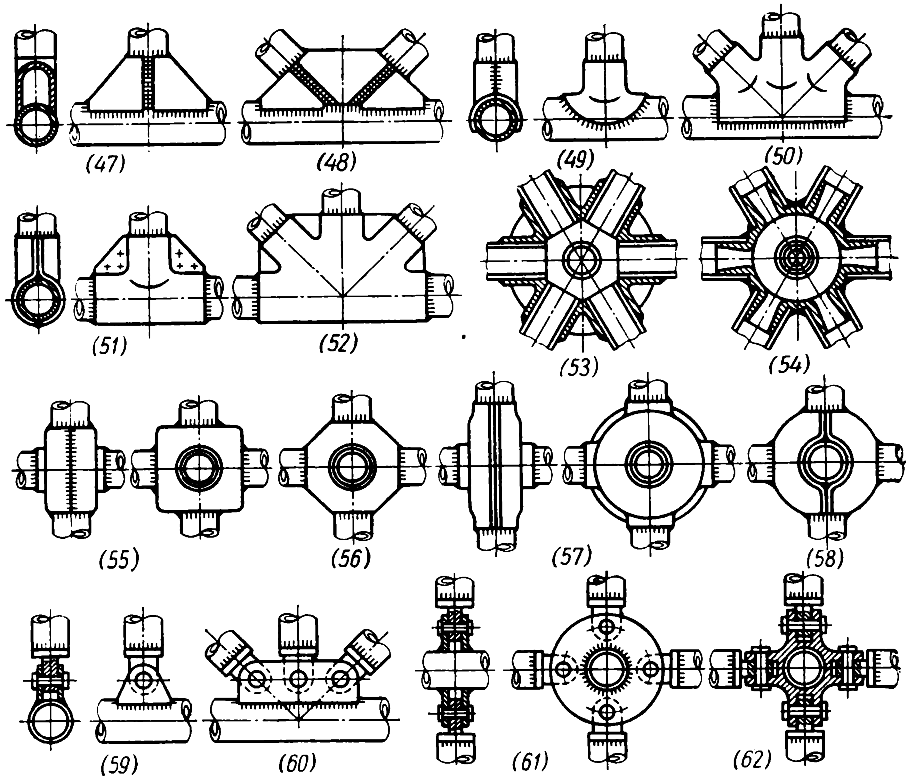


Fig. 204 (continued)

Welding in sleeves made of solid (Fig. 204, *31-33*) or welded (Fig. 240, *34*) tubes is also employed.

Very often tube joints are strengthened by gusset plates which are butt-welded (Fig. 204, *35*, *36*), butt- and slot-welded on one of the tubes (Fig. 204, *37*, *38*) and slot-welded over all tubes being connected (Fig. 204, *39*, *40*).

Slot-connection by gusset plates and with preparation of the ends of tubes in a hot state (Fig. 204, *41*, *42*) makes it possible to join several tubes in one unit, and is employed in multi-ray units. The shortcomings include low rigidity in the plane of the gusset plates and difficult preparation of the tubes.

Rigidity can be increased with the aid of double gusset plates (Fig. 204, *43*, *44*) but the distance between the plates (in the direction normal to their plane) should be selected so that the edges of the adjacent plates can be made by one weld *m* (Fig. 204, *46*, *47*).

U-shaped gusset plates (Fig. 204, *46*, *48*) are strongest and most rigid.

Heavily loaded units employ joints with pressed straps enveloping the tubes (Fig. 204, *49*, *50*). The rigidity of the joint can be increased if the straps are provided with gusset plates joined by spot welding (Fig. 204, *51*, *52*).

In multi-ray joints tubes are welded to star-shaped forged pieces with recesses (Fig. 204, *53*) or with necks (Fig. 204, *54*) for the tubes. Multi-ray units are also connected with the aid of prismatic (Fig. 204, *55*, *56*), cylindrical (Fig. 204, *57*) or spherical welding boxes (Fig. 204, *58*). The latter method can be used to join tubes practically at any spatial angle.

Figure 204, *59-62* illustrates examples of hinged connection of welded tubes in truss units.

Chapter 6

Riveted Joints

In the past, riveting was the principal method of connecting structures made of sheets and plates (reservoirs, boilers, etc.) as well as frames and trusses made from shaped rolled stock. Today, rivets have been almost completely replaced in this field by welded structures which are stronger and more effective.

Rivets are employed for:

joints where it is necessary to preclude the thermal aftereffects of welding which deteriorate the metal structure in the weld area, overheat the parts close to the welded joint and warp the products;

joints made of metals with poor weldability and heterogeneous metal joints (for example, steel and nonferrous alloys, etc.);

joints of metal elements with nonmetallic materials (wood, leather, fabrics and plastics which cannot be fastened by pressing, bonding, etc.).

Up till now rivets are the main kind of fasteners in light frames and thin-sheet shells made of light alloys (especially in the aircraft industry). This is due to the fact that light alloys are difficult to weld, the welded joints have a low vibrational resistance, and the inevitable warping especially pronounced when long products are welded. Intricate forms and restricted overall dimensions inherent in aircraft designs make it difficult to manipulate welding devices and check the quality of welded joints.

6.1. Hot Riveting

Hot riveting is employed in power and strong-tight joints when the diameter of the rivets is over 8-10 mm. Rivets of a smaller diameter are as a rule inserted by the cold method.

A rivet with a *set head* is heated to a plastic state (900-1,000°C) and its shank is inserted into a drilled hole in the members to be fastened after which, while holding its head, the protruding shank end is upset by an impact or pressing tool (Fig. 205*a*) to form a second, *closing* head (Fig. 205*b*). As it cools the rivet contracts in length and tightly compresses the members being joined.

The strength of the joint is almost entirely determined by the forces of friction arising on the abutting surface of the parts due to the shrinkage of the rivet.

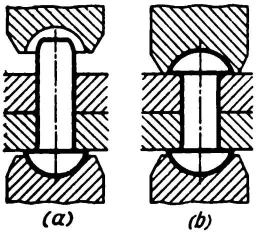


Fig. 205. Hot riveting

At the initial stage of cooling when the metal of the rivet is in plastic state, its shank is elongated and its diameter is reduced. At this time the rivet does not produce any appreciable pressure on the members being connected. As the temperature drops, the material of the rivet becomes stronger and offers resistance to shrinkage. The final compressing force is determined by the shortening of the rivet during the cooling period from the temperature at which the plastic deformations of the rivet material give way to elastic ones down to the temperature of complete cooling. This shortening also determines the magnitude of tensile stresses in the rivet shank.

During the cooling process the diameter of the shank is diminished due to plastic elongation at the initial period of cooling, due to elastic elongation and reduction in the transverse dimensions upon final cooling. The volume of a rivet also changes because of the γ-α-transformation occurring during cooling.

The conjoint effect of these factors forms a clearance amounting to tenths of fractions of a millimetre between the shank and the walls of the hole even if the rivet is initially introduced into the hole with a push fit, using a hammer for example.

The method commonly used today for calculating riveted joints for shear of the shanks and crushing of the walls of a hole and the

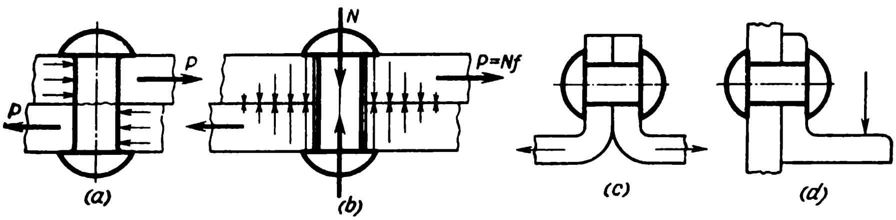


Fig. 206. Calculation of rivets

surfaces of shanks under the action of tensile force P (Fig. 206*a*) disagrees with the actual working conditions of riveted joints.

Rivets are subjected to shear only after the members being connected are offset by the amount of clearance between the shank of the rivet and the walls of the hole, i.e., when the destruction of a riveted joint commences.

When calculating hot-riveted joints it is more correct to accept as the basis the axial force N developed by a rivet during shrinkage, and the friction force $P = Nf$ in the joint (Fig. 206*b*). The axial force is

$$N = \sigma F$$

where F = cross-sectional area of the rivet
σ = tensile stress produced in the rivet at the end of shrinkage

$$\sigma = E\alpha\,(t_1 - t_0)$$

Here E and α = the modulus of normal elasticity and the coefficient of linear expansion of the rivet material, respectively
t_0 = final cooling temperature
t_1 = the temperature at which plastic yield of the rivet material ceases and elastic elongation of the rivet shank begins

The difficulty of calculation by this method consists in the fact that the values in the equation are variable. The values of E and α depend on the temperature, and the temperature t_1 is uncertain because the period of transition of plastic deformations into elastic ones is prolonged. The calculation is also complicated due to unequal heating of the rivets before riveting and also due to an unequal temperature range along the axis of the rivets. For example, only the free end of a rivet is frequently heated to form the closing head while the set head is left cold. In this case the compressive force is considerably decreased.

Pure shear (Fig. 206*a*, *b*) seldom occurs in practice. In most cases riveted joints are subjected to additional stresses, for example, to bending or tension (Fig. 206*c*, *d*) caused by deformation of the unit under the action of external forces.

The calculation in common use disregards the decisive factor of strength—the tension of a rivet due to contraction during cooling. Even if the functioning of rivets for shear were taken as the basis the calculation would have to be conducted according to a combined stressed state of shear and tension.

The parameters of riveted joints are selected in practice from designs of available structures accounting at the same time for the specific operating conditions of the joint (requirements of tightness, working temperatures, effect of aggressive media, etc.). Almost every field of application of hot riveted joints has its own standards against which the joint is checked in operation.

6.2. Cold Riveting

In the case of cold riveting contraction of a rivet is caused only by plastic deformation of its material during closing up. With the cold method the axial force compressing the members being connected is less than in hot riveting and depends on the degree of plastic deformation of the rivets which may vary within wide range and has more or less constant magnitude only in machine riveting, for example hydraulic riveting.

Unlike hot-riveted joints, the strength of cold-formed joints is mainly determined by the resistance of rivets to *shear*. The forces of friction in the joint relieve the rivets of shear and compression.

The basic problem when designing cold-riveted joints is to ensure correct functioning of rivets in shear primarily by fitting the rivets without clearance into holes. In critical joints the holes for rivets in the members to be connected should always be machined simultaneously. The rivets should be driven into holes with interference (for which purpose it is necessary in most cases to accurately machine both the holes and the rivet shanks). When rivets are fitted with a clearance the plastic deformation should be enough to clamp the members together and assure spreading the shank into the clearance

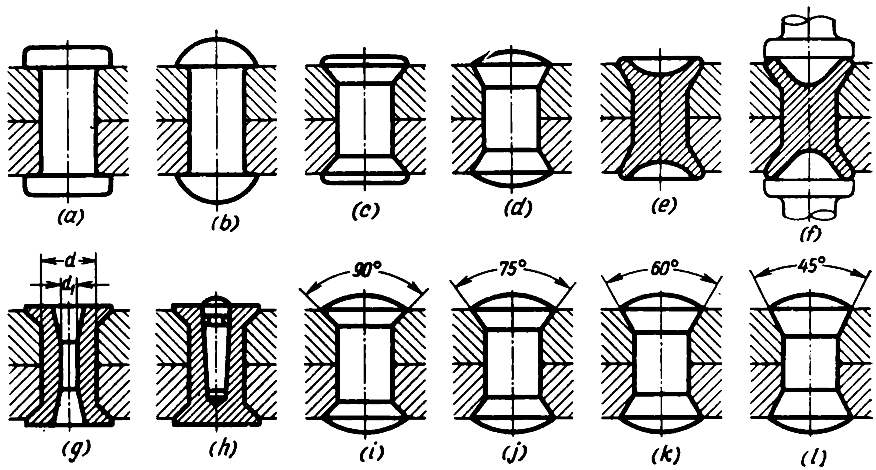


Fig. 207. Varieties of rivets

tightly and fit the shank against the walls of the hole, especially in the abutting plane of the elements being connected. It is more advantageous to employ rivets not with flat, button and similar heads (Fig. 207*a*, *b*) resting against the surfaces of the jointed members but rivets with countersunk heads (Fig. 207*c*-*f*) when the clinching force affects in a large measure the shank expanding it transversely. In this respect the best design of a rivet is the one with a neck in the plane of shear compacted by snaps on both sides (Fig. 207*f*).

The other useful designs are hollow rivets, for example with a thickening in the plane of shear expanded by a punch after the rivet is driven in (Fig. 207*g*). In hollow rivets the reduction in the cross section subjected to shear which is generally very small when $\frac{d_1}{d} < 0.5$ (d_1—diameter of the internal hole in the rivet, d—rivet diameter) can be eliminated by means of a plug-type punch left in the rivet (Fig. 207*h*).

In the case of cold riveting it is good practice to use round-top countersunk rivets with an angle of 75-60° and even 45° (Fig. 207*j-l*) to facilitate spreading of the shank during upsetting.

In hot riveting preference is given to heads with a flat bearing surface or a bevel angle of over 75° (Fig. 207*i*, *j*). When the angles are small high compressive and rupturing stresses develop in the countersunk area in the riveted members, whereas the clamping force diminishes.

When rivets are clinched cold the strength of the joint is favourably affected by the cold working of a rivet due to the closing up force, which strengthens the material of the rivet.

In mechanical engineering, cold riveting is the method usually preferred because riveted joints are mainly intended to eliminate thermal aftereffects and obtain strong joints between parts without impairing the accuracy of their dimensions and mutual arrangement.

Rivets are used, for example, to fasten counterweights to the webs of crankshafts, the rims of gear to disks, lining plates to massive parts, friction linings to clutch disks and brake shoes. Rivets are also employed to connect light sheet structures such as pressed cages for ball bearings.

The absence of thermal aftereffects, simple design and high efficiency make cold riveting superior in many cases to the hot method even when plates and parts of large cross section are joined.

Cold riveting is not practicable for joints intended to operate at high temperatures since such temperatures take off the cold working and diminish the force of compression produced during riveting.

6.3. Rivet Materials

For general-purpose hot-riveted joints designers employ rivets made of carbon steel 30, 35 and 45. Rivets for special joints are manufactured from stainless steel and heat-resistant alloys to suit the conditions of operation.

Rivets for joining steel parts by the cold method are made of soft steel grades 10 and 20 and in important joints of steel 15X and 20X (Soviet standard specifications) which are plastic and have higher strength.

Nonferrous metals are connected and soft materials are joined to metal parts by means of copper, brass, bronze, aluminium and aluminium-alloy rivets. With higher requirements for corrosion resistance, the rivets are made of stainless steel, Monel metal, nickel and titanium alloys.

Power joints made of aluminium alloys are connected by duralumin rivets

Using the ageing property of duralumin rivets are driven in a freshly quenched state (quenching in water from the temperature of 500-520°C) when the material of the rivets retains plasticity for 0.5-2 hours after the quenching

process. After four or six days at $t = 20°C$ (*natural ageing*) the material of the rivets ages and acquires increased strength and hardness. *Artificial ageing* (at 150-175°C) reduces the ageing process to 1-4 hours.

In mass production large lots of quenched rivets are kept in refrigerating chambers at a minus temperature (about —50°C) which delays ageing practically for an unlimited period of time. Some deformable alloys Д3П, Д18П, В65, В94 (Soviet standard specifications) possess good plasticity after ageing and can be riveted in an aged state.

Metals with different electrochemical potentials are not recommended for riveted joints. They form galvanic pairs and accelerate the process of corrosion. As a rule, rivets are made of the same material as the parts being joined.

In joints with heterogeneous metals (for example, aluminium rivets in parts made of magnesium and copper alloys) the rivets should be coated with cadmium or zinc for protection.

6.4. Types of Riveted Joints

A riveted joint should be loaded only in shear and relieved of the action of bending moments which cause unilateral bending of the rivet shanks. The rupturing stresses developing in bending are added to the tensile stresses which appear during riveting and overload the shank and the head of the rivet.

In the joints in Fig. 208*a*, *b* the tensile forces produce a bending moment approximately equal to the product of the tensile force by the thickness of the material (Fig. 208*h*, *i*). This moment is partly damped by the resistance to the elastic bending of the plates, and is partly transmitted to the rivets.

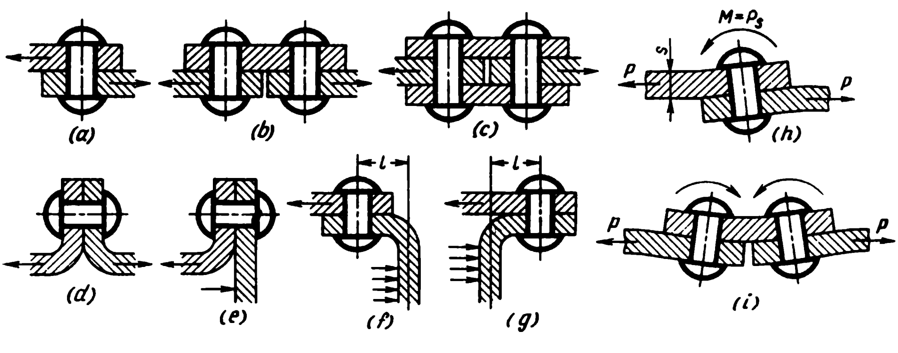


Fig. 208. Riveted joints

In a two-strap joint (Fig. 208*c*) the central application of forces prevents a bending moment. Besides, this joint is in *double shear* and due to the double friction surfaces the resistance to shear in this case is twice as large as in the designs *a* and *b*.

The design in Fig. 208*d* with flanged edges is irrational since the rivets are bent in tension.

In corner joints (Fig. 208*e*) with one flanged edge which are sometimes used to connect bottoms to the shells of reservoirs containing pressurized gases or liquids, deformation of the walls of the reservoirs causes the rivets to bend.

Joints in Fig. 208*f*, *g* where the rivets are mainly subjected to shear and only to a slight degree to bending are more practicable. The smaller the deformation (caused by the internal pressure) of the shell bottom and walls closest to the seam, i.e., the smaller the distance l between the rivet and the bottom surface, the smaller the bending.

Single-row (Fig. 209*a*, *d*), double-row (Fig. 209*b*, *e*) and multiple-row (Fig. 209*c*) riveted joints are used. In double and multiple

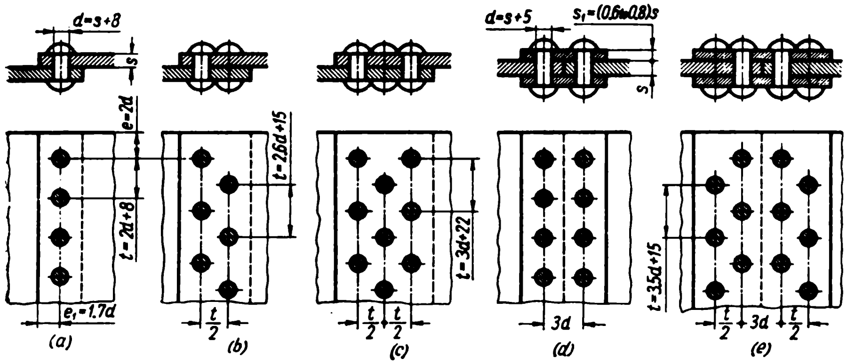


Fig. 209. Parameters of riveted joints

seams the rivets are usually arranged in staggered order to achieve more uniform loading of the seams and make it easier to drive in the rivets.

Figure 209 shows the empirical relations (for general-purpose structures) between rivet diameter d and pitch t on the one hand and distances e and e_1, on the other.

Due to the weakening effect of holes the strength of riveted joints is less than that of the solid material.

The relative strength of joints as expressed in fractions of strength of solid material is presented in the table below.

Type of joint	Seam		
	single-row	double-row	triple-row
Lap joint (Fig. 209*b*)	0.5-0.6	0.6-0.7	0.7-0.8
Butt joint (Fig. 209*e*)	0.6-0.7	0.75-0.85	0.85-0.9

An increase in the number of rows above three only slightly increases the strength.

By their function, distinction is made between the strong seams employed in power structures, and *strong-tight* seams which accept forces and ensure good joint tightness. The latter are employed, therefore, for all kinds of reservoirs. Strong-tight seams are constructed with rivets having reinforced heads, usually with conical underheads which make a rivet fit tightly in its hole. Rivets in the strong-tight joints functioning at high temperatures are hot-driven irrespective of the thickness of the parts being joined. The joints usually have two or three rows of rivets.

The tightness of a joint is attained by additional means, for example by applying sealing compounds (red lead dissolved in oil, greases based on synthetic resins, etc.) to the surfaces of the joint before riveting. It should be remembered, however, that sealing greases reduce the coefficient of friction at the joint surface and the shear strength of the joint. For this reason, the grease should be applied not over the entire surface of the joint but in a narrow band meandering around the holes for the rivets.

For joints operating at high temperatures siloxane enamels are used with metallic powders (Al, Zn) which endure a temperature of up to 600°C.

Another method of sealing is to provide the joint with thin soft metal wires which flatten during the process of riveting.

Good results are obtained when the surfaces of a joint, cleaned in advance, are coated with plastic metals applied by the galvanic method or gas-flame pulverization. Copper and nickel coatings are the most thermostable.

Metallic coatings increase the strength of a joint since the high temperature and pressure on the surface of the joint produce mutual diffusion of the coating metals with the formation of an intermediate metal structure layer.

Sometimes the edges of a joint are calked (Fig. 210*a*) at an angle of 15-20°.

Calked seams will retain their tightness in operation only if the joint is sufficiently rigid. When the joint has insufficient rigidity its sealing especially under cyclic loads is quickly destroyed as a result of periodic deformations ("breathing").

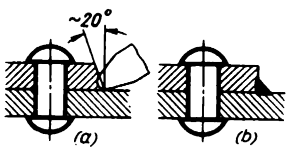


Fig. 210. Calking (*a*) and welding up (*b*) of edges

The method of a joint tightening by securing the edges of the joint by a light weld is sometimes used (Fig. 210*b*) but must not be accepted as rational. The rigidity of welds, even of small cross section, is significantly larger than that of riveted seams. For this reason a weld accepts the load acting on the joint. The strength of the seam determines the strength of a joint. In such cases it is better to join the parts with a weld of normal cross section.

6.5. Types of Rivets

Table 13 illustrates the types of rivets for strong (Table 13, *1-6*) and strong-tight (Table 13, *7-13*) joints.

The range of rivets for strong-tight joints includes rivets with a conical underhead (Table 13, *9*) which ensure a tight fit of the rivet.

Pan-head rivets (Table 13, *8*, *9*) are intended for joints subjected to the action of hot gases (fireboxes, flues, etc.) assuming that such heads resist hot erosion longer and retain their strength even with considerable burn-out.

However, at an increased temperature, especially under the action of a gas flux, round-top (Table 13, *10*) or flat-top (Table 13, *11*) countersunk rivets are more advantageous. Rivets made of heat-resistant alloys are more useful in these conditions.

Sketches *14-19* show small rivets made of nonferrous metals and rivets for tin-smith and copper-smith work.

Rivets are designated on drawings and in technical documents by the number of the USSR State Standards, diameter d of the shank and rivet length l as specified in appropriate USSR State Standards.

Example:

Rivet ГОСТ 1187-41 10·30.

In the case of nonstandard rivets it is necessary to present complete drawings of the rivet and riveted joints specifying the material, the kind of processing, the accuracy of manufacture and the needed technical requirements.

6.6. Design Relative Proportions

Figure 211*a* illustrates design proportions of rivet set heads most widely used for strong and strong-tight joints.

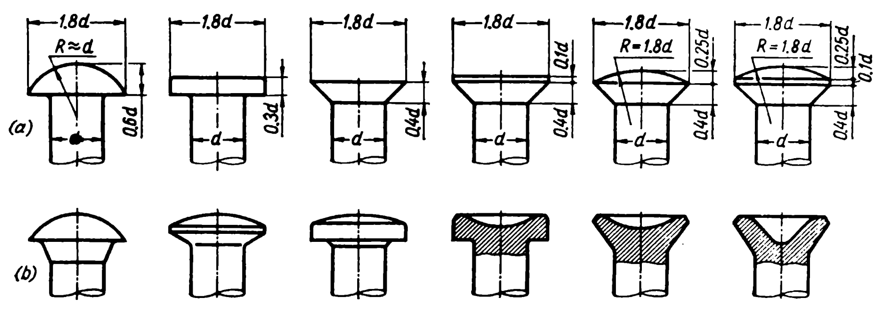


Fig. 211. Shapes of rivet heads

Clinched heads usually resemble set heads. The other forms of clinched heads are shown in Fig. 211*b*.

The rivet diameter cannot be selected by only one rule. It depends on the thickness of the materials being connected, the spacing pitch

Table 13

Types of Rivets

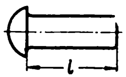
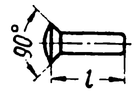
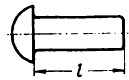
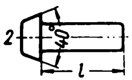
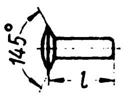
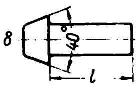
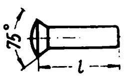
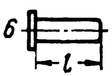
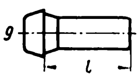
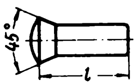
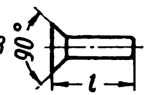
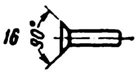
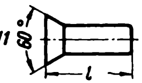
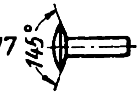
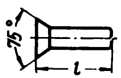
Sketch of design	USSR State Standard: rivet diameter, mm	Sketch of design	USSR State Standard; rivet diameter, mm	Sketch of design	USSR State Standard: rivet diameter, mm
1 	1187-11; 1-37	4 	1192-41; 6-8	7 	1191-41; 8-37
2 	1188-41; 6-34	5 	1190-41; 2-7	8 	1193-41; 6-34
3 	1192-41; 10-13	6 	1189-41; 2.3-6	9 	1194-41; 6-34

Table 13 (continued)

Sketch of design	USSR State Standard; rivet diameter, mm	Sketch of design	USSR State Standard; rivet diameter, mm	Sketch of design	USSR State Standard; rivet diameter, mm
10 	1192-41; 1195-41; 28-37	13 	1192-41; 1195-41; 1-8	16 	ОСТ НКТ 8218/1170; 2-9
11 	1192-41; 16-25	14 		17 	
12 	1192-41; 10-13	15 		18 	
				19 	

of rivets, the type and magnitude of load, the relationship between the strength and hardness of the materials of the rivet and the parts being joined and, finally, on the method used to drive in the rivet.

If we proceed from the functioning of a rivet in shear and base our calculations on the condition of equal strength of the rivets (in shear and compression) and of the riveted plates (in compression, shear and rupture at the critical sections), then for the particular case of a single-row lap joint (Fig. 212*a*) with the same strength of the material of the rivets and the plates the following relationships can be obtained:

$$d = 2s; \; t = 2.5d; \; e = 1.5d$$

This calculation gives exaggerated values of rivet diameter (especially when the values of s are large) and reduced values of the pitch.

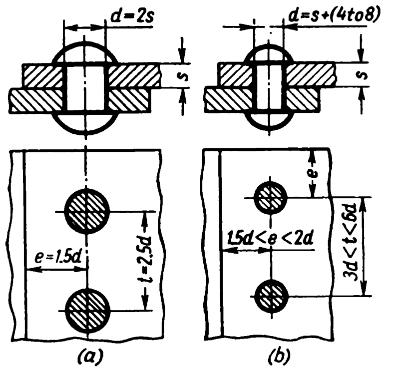


Fig. 212. Design proportions of riveted joints

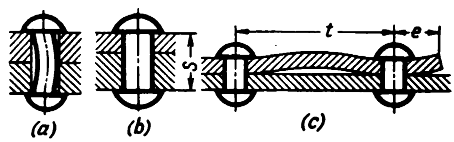


Fig. 213. Determining the diameter and spacing pitch of rivets

In practice, use is made of the following relationships (Fig. 212*b*):

$$d = s + (4\text{-}8) \tag{6}$$
$$3d < t < 6d \tag{7}$$
$$1.5d < e < 2d \tag{8}$$

In these formulas all dimensions are in millimetres.

Rivets with diameters smaller than those determined from formula (6) are difficult to forge and they may bend in the hole (Fig. 213*a*). Clinching of large-diameter rivets may overstress the material of the members being connected.

When materials of various thickness are riveted it is necessary to take as the basis their total thickness S (Fig. 213*b*). When S = 5-60 mm the diameter of a rivet can be found from the formula

$$d = (3\text{-}3.5)\sqrt{S} \text{ mm}$$

The pitch of rivets should never exceed $6d$, otherwise the tightness of the joint sections between the rivets may be impaired (Fig. 213*c*). When $t < 3d$ it is difficult to fit the rivets.

The length of edge e should not exceed $2d$ because the edge is liable to curve off (Fig. 213*c*). If $e < 1.5\,d$ the edge can be damaged when clinching the rivets. Relatively small and closely located rivets should be preferred to large and spaced ones, i.e., the lower limits should be selected in formulas (6) and (7).

These are tentative relationships. It is better to rely on the experience derived from available structures and employ the standards approved for a given branch of industry, and conduct experimental verification when designing new constructions.

In the case of cold-driven rivets the shear calculation is more than substantiated. But here too there are factors difficult to account for (for example, the magnitude of the force applied to the rivet and the degree of plastic deformation which determines the fit of a rivet in the hole). The admissible stresses are assumed to be equal to the ultimate strength of the rivet material in shear and compression with the factor of safety equal to 3-4. Besides, the mode of processing the holes is taken into account.

The design stresses in kgf/mm² for the rivets made of steel 10 and 20 (USSR State Standards) are given in the table below.

Load	Punched holes	Drilled holes
Shear	10	15
Compression	20	30

In the case of pulsating load the admissible stresses are reduced by 10-20 per cent and with a load variable in direction by 30-50 per cent.

6.7. Heading Allowances

Let F be half of the cross-sectional area of the rivet head minus the cross section of the rivet body (hachured area in Fig. 214a). The volume of this portion of the head is equal to

$$V = F\pi d_{c \cdot g}$$

where $d_{c \cdot g}$ = diameter of location of the centre of gravity of this area.

The height h of the allowance necessary to fill this volume can be determined from the formula

$$V = F\pi d_{c \cdot g} = \frac{\pi d^2}{4} h$$

where d = rivet body diameter.

Hence,

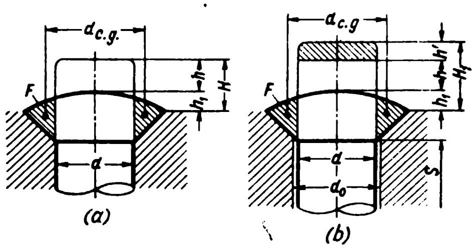


Fig. 214. Determining the heading allowances

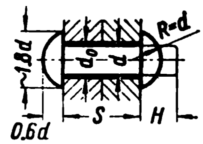
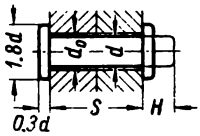
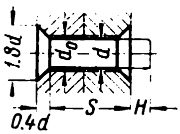
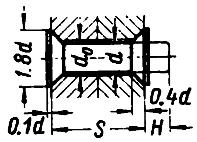
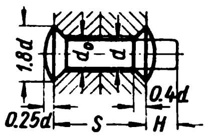
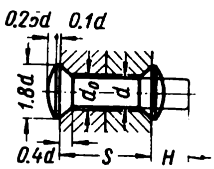
$$h = \frac{4Fd_{c \cdot g}}{d^2}$$

The height H of the allowance above the surface of the riveted member is

$$H = h + h_1 = \frac{4Fd_{c \cdot g}}{d^2} + h_1 \tag{9}$$

Table 14

Rivet Heading Allowances

Sketch	Allowance: for rivets driven without clearance	Allowance: for rivets driven with clearance
	$H=1.2d$	$H'=1.2d+S\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx 1.2d+0.1S$
	$H=d$	$H'=d+S\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx d+0.1S$
	$H=0.6d$	$H'=0.6d+(S-0.8d)\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx 0.5d+0.1S$
	$H=0.8d$	$H'=0.8d+(S-0.8d)\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx 0.7d+0.1S$
	$H=d$	$H'=d+(S-0.8d)\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx 0.9d+0.1S$
	$H=1.2d$	$H'=1.2d+(S-0.8d)\left[\left(\frac{d_0}{d}\right)^2-1\right]\approx$ $\approx 1.1d+0.1S$

where h_1 = height of the upset rivet head depending on the shape of the head.

Formula (9) is valid for rivets driven into holes without clearance.

For rivets fitted into holes with a clearance which is filled up during clinching account should be taken of the penetration of the metal into the annular space between the hole and the rivet shank amounting to

$$V' = \frac{\pi}{4}(d_0^2 - d^2)\,S$$

where d_0 = diameter and S = length of the hole (Fig. 214*b*).

The additional height of allowance h' can be found from the formula

$$V' = \frac{\pi}{4}(d_0 - d^2)\,S = \frac{\pi d^2}{4}h'$$

whence

$$h' = S\left[\left(\frac{d_0}{d}\right)^2 - 1\right]$$

The total height of the allowance is

$$H' = H + h' = H + S\left[\left(\frac{d_0}{d}\right)^2 - 1\right] \tag{10}$$

Account should be taken of the manufacturing tolerance for holes and rivets (regular hot upset rivets are made to the 5-7th classes of accuracy and cold upset rivets to the 3-4th classes), introducing into the calculations the maximum diameter of the hole and the minimum diameter of the rivet.

In hot riveting, consider also the increase of the rivet diameter when it is heated. The diameter of a heated rivet is

$$d = d_0(1 + \alpha t)$$

where d_0 = diameter of a cold rivet
α = coefficient of linear expansion of the rivet material (for steel $\alpha \approx 11 \times 10^{-6}$)
t = increase in temperature

If for average conditions $\frac{d_0}{d} = 1.05$, then

$$H' \approx H + 0.1S \tag{11}$$

Table 14 illustrates the values of H calculated on the basis of formula (9), and H' on the basis of formulas (10-11) for the most common types of rivets.

The length of rivets with the allowance calculated by formulas (9-11) should be rounded off to the nearest larger standard length.

6.8. Design Rules

The holes for rivets in the members to be fastened should be machined simultaneously. Misaligned holes (Fig. 215*a*) significantly weaken the rivet.

The holes in critical joints should be reamed together and the rivet driven in with interference (Fig. 215*b*).

Arrangement of rivets in constricted places should be avoided (Fig. 215*c*). The space around rivets should be wide enough to admit

a riveting tool. The distance e (Fig. 215d) from the rivet axis to the nearest vertical walls and other elements of the structure which may interfere with the approach of the riveting tool should not be less than (2-2.5) d when pneumatically riveting and (1.5-2) d when hydraulically riveting. The minimum distance from the edge $e_1 = 1.7d$.

It is especially important to provide free access to the closing head. When riveting profiled pieces place the closing head in an open space (Fig. 215e). The design in Fig. 215f is wrong.

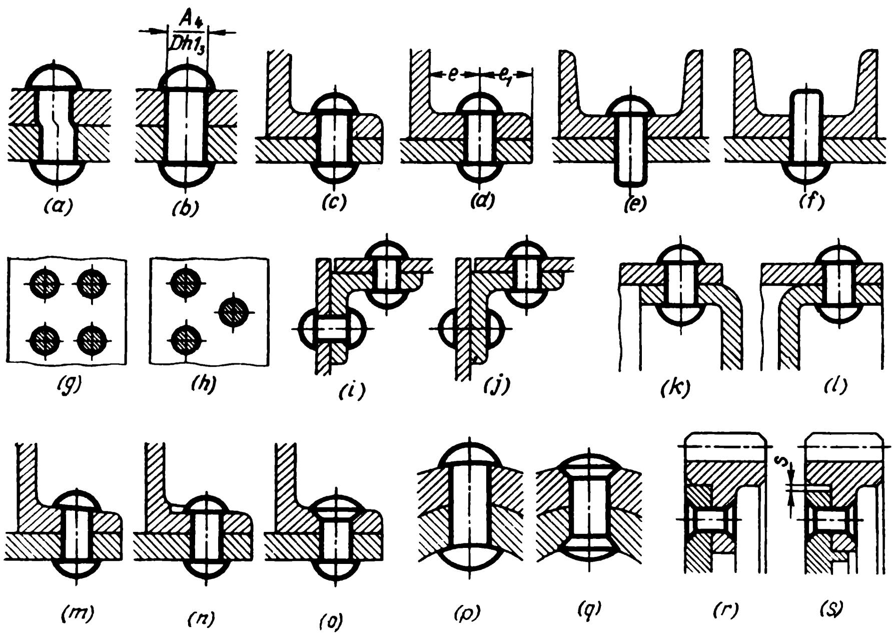


Fig. 215. Installation of rivets

In adjacent seams with parallel (Fig. 215d, g, h) or perpendicular (Fig. 215i, j) arrangement of rivet axes stagger the rivets to facilitate riveting (Fig. 215h, j).

The distance from the axes of the rivets to the extreme edges of the parts being joined should be reduced to the minimum so that the use of cumbersome riveting tools with a large outreach may be avoided. Thus, when connecting bottoms of cylindrical reservoirs to shells the outward flanges (Fig. 215l) should be preferred to the inward ones (Fig. 215k) although the former design is inferior in strength.

When riveting on inclined surfaces (Fig. 215m) it is necessary to use hot riveting with heating the entire rivet, flats made on the inclined surfaces (Fig. 215n) or else countersunk rivets (Fig. 215o).

The same rule applies to rivets mounted on cylindrical surfaces (Fig. 215*p*, *q*).

When cold riveting parts which are to preserve their accurate dimensions (for example, when the rims of gears are riveted to disks, Fig. 215*r*), consider possible deformation of the walls under the action of the riveting forces (especially with rivets having countersunk heads). The sections of the material deformed by riveting should be spaced from the accurate surfaces by a clearance (*s* in Fig. 215*s*).

To prevent deformation of the members being fastened rivets are also clinched by a carefully regulated hydraulic force.

6.9. Strengthening of Riveted Joints

Apart from a proper selection of geometrical parameters (diameter, number of rows of rivets, pitch of spacing rivets) the strength of riveted joints can be increased by certain manufacturing measures.

Rivets are most commonly made of alloy steel 40X (USSR State Standards). If an undriven rivet is heated to a temperature exceeding the phase transformation temperature, i.e., up to 750-800°C and cooled rapidly enough the steel is mildly quenched to a sorbite structure, which significantly increases the strength of the joint. Thus if rivets are made from alloy steel they can be appreciably strengthened by the transformation which occurs during cooling.

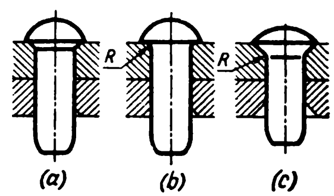


Fig. 216. Shapes of rivets and holes

For very important joints the use of rivets made of high-strength martensite-ageing steel which strengthens in the course of cooling is of good practical value.

To prevent a coarse-grained metal structure rivets must not be heated above 1,000°C.

The cyclic strength of riveted joints can considerably be enhanced if the holes for rivets are properly machined and the rivets are most rationally designed. Punching of holes should be avoided because this may cause tears and microcracks, which become the sources of considerable stress concentration on the edges of the holes. Holes for rivets should be drilled (simultaneously in the parts being joined), reamed or when cold riveting reamed and broached.

The edges of inlet and outlet sections of holes should be chamfered (Fig. 216*a*) or still better radiused (Fig. 216*b*, *c*), and the surfaces of these sections in the case of cold riveting should also be pressed.

The set heads of rivets should be connected to the shank by smooth fillets or at least by chamfers (conforming to the fillet or chamfer dimensions at the hole edges).

When hot riveting the head-to-shank transitions are self-forming as the metal fills in the chamfer or fillet of the hole.

The strength of a joint can be improved by increasing the adhesion between the contacting surfaces. It is expedient to shot blast the abutting surfaces to increase their roughness or to fine rifle them. In such cases the abutting surfaces should be metallized to ensure a proper seal.

An effective method of increasing the strength of hot-riveted joints is hydraulic clinching when the rivet and the joined members are held by a constant force until the rivet cools down.

The heated rivet is introduced into the hole and compressed by a heavy force to form the closing head and expand the shank until it tightly fits the hole. The rivet is held compressed until it cools down to 200-300°C. The reduction in the diameter of the rivet shank during cooling is compensated for by the continuous compression of the rivet by the set punch.

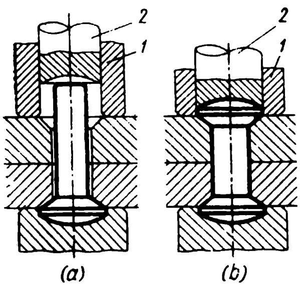


Fig. 217. Riveting with a double-action set punch

Consequently a joint is obtained with a rivet fitted practically without clearance and reliably secured against shear. The joint also preserves an increased resistance to shear typical of hot-riveted joints due to the forces of friction in the joint developing at the initial stage of the process when the joint is compressed by the set punch, and at the final stage due to the axial contraction of the rivet shank when cooling from the final temperature of riveting to the temperature of the ambient air.

The process of hydraulic rivet closing up requires a higher riveting pressure sufficient to deform the shank in a semi-plastic state (at temperatures corresponding to the final period of riveting). This process is less productive (due to the prolonged holding) than the usual process. However, this is justified by a high quality of the joint.

The use of a double-action set punch to separate the process of compressing the joint from the process of plastic deformation of the rivet will undoubtedly increase the strength of the joint.

In this case the members to be connected are first compressed with the aid of an external annular punch *1* (Fig. 217*a*) and then an internal set punch *2* is used to apply a force to the shank of the rivet to form the closing head and expand the shank until the initial clearance between the shank and the hole is completely eliminated (Fig. 217*b*).

The entire system is held in this state until the rivet cools. As in the previous case, the shrinkage of the rivet shank in the axial direction when cooling is compensated by the plastic deformation of the rivet under the action of the set punch. This is the reason why countersunk heads are always preferred. After the rivet is cooled the pressure is taken off the set punch *2* and then after a certain delay off the punch *1*. The joint is clamped tight by the contraction of the rivet shank upon full cooling, the process occurring in the elastic stage.

6.10. Solid Rivets

These rivets (Fig. 218) are employed for heavy loaded joints. The stud of such a rivet is made of strong heat-treated steel fitted into the hole with interference. Since the stud cannot be clinched

the closing head is formed by rolling rings made of plastic metal into annular grooves on the bar.

The amount of axial interference depends on the design of the closing head. The joints in Fig. 218*a* and *b* serve only to hold the stud in place. Greater interferences can be obtained with the increase

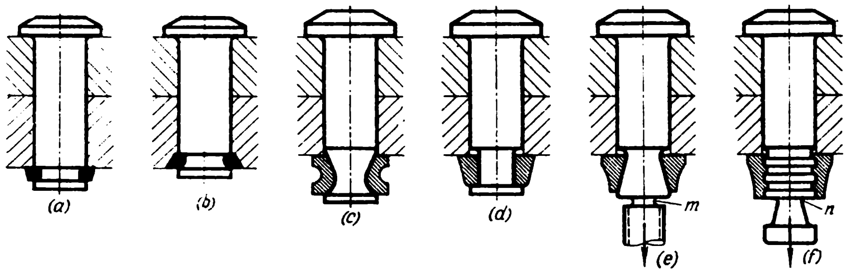


Fig. 218. Solid rivets

of the height of the heads (Fig. 218*c*, *d*). The rings are fixed fastened by a circular rolling (Fig. 218*c*) or with the aid of a tapered snap (Fig. 218*d*) or else by expandable snaps.

In the designs in Fig. 218*c*, *d* the head of the stud has to be supported while the ring is snapped. If the access to the head is difficult, use is made of the designs in Fig. 218*e*, *f*. During the process of snapping the joint is clamped by the thread (Fig. 218*e*) or the head (Fig. 218*f*) with the clamping force applied against the surface of the clamped parts. After the closure of the rivet the threaded bar (or head) is broken off at the thin necks *m* and *n*.

6.11. Tubular Rivets

These rivets are used to fasten joints carrying small loads.

The rivets are made from profiled tubes. The set head is usually preformed (Fig. 219*a*). The other end of the rivet is expanded by means of a punch (Fig. 219*b*) and in the case of large-diameter rivets with the aid of revolving rolls. The revolving tool is also employed to rivet parts made of brittle materials.

Sometimes, especially in the case of countersunk installation both rivet heads are formed simultaneously by set punches acting on two sides (Fig. 219*c*, *d*).

All types of tubular rivets are amenable to additional internal expansion which increases the tightness of the shank fit in the hole and improves the shear strength of the joint. In designs with a preformed head the expansion can be done at the same time as the closing head is formed by the protruding set punch (Fig. 219*e*).

Tubular rivets can be reinforced by press-fitted studs. The studs are secured by rifles (Fig. 219*f*), annular grooves and by calking-in the end-faces.

If the surface of a joint must be smooth (for example, when lining sheets are attached by rivets), use is made of *semi-tubular rivets* with flat (Fig. 219*g*, *h*), countersunk (Fig. 219*i*) or button (Fig. 219*j*)

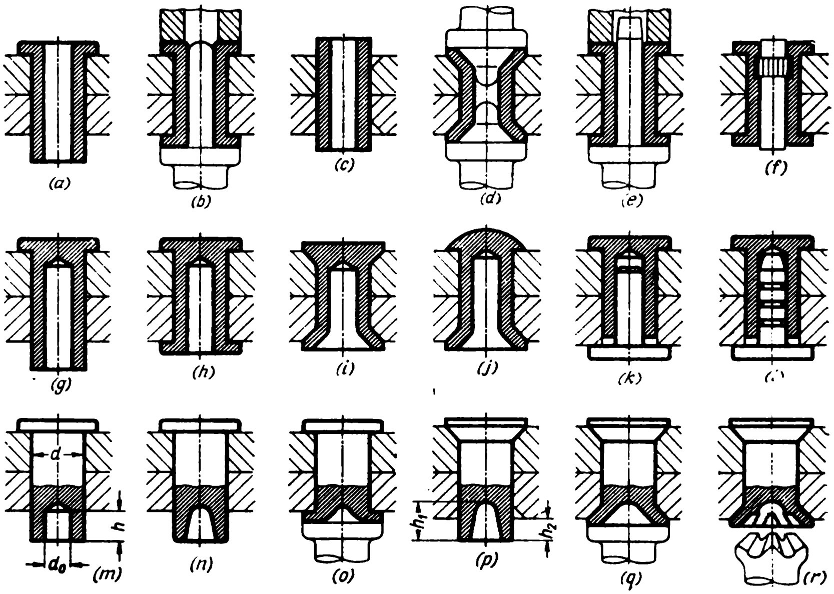


Fig. 219. Tubular and semi-tubular rivets

heads. If a smooth surface on both sides is required, a capped stud is press-fitted into the rivet (Fig. 219*k*). The stud is held in the rivet by friction forces. Annular grooves on the stud (Fig. 219*l*) make the engagement tighter.

Rivets with undercut ends (Fig. 219*m*) possess an increased shear resistance. Undercutting is performed by drilling (Fig. 219*m*) or punching (Fig. 219*n*). The diameter of undercut is usually $d_0 = (0.5\text{-}0.6)\ d$ (where d is the diameter of the rivet). The depth of undercut (and the height of the protruding end of the rivet) when set punching onto a surface (Fig. 219*m*) is $h = (0.5\text{-}0.6)\ d$. In the case of countersunk rivets (Fig. 219*p*) the depth of undercut $h_1 = (0.6\text{-}0.7)\ d$ and the height of protrusion $h_2 = (0.3\text{-}0.4)\ d$.

As in the previous cases the end of the rivet is upset by a set punch (Fig. 219*o*, *q*). Star-like set punches (Fig. 219*r*) are used in light joints to reduce the upsetting force.

6.12. Thin-Walled Tubular Rivets

These rivets are manufactured from thin-walled (0.2-0.5 mm) tubes and are usually employed to fasten soft materials (leather, fabrics, plastics, etc.).

The simplest type of such a rivet is a tube expanded on both sides onto a surface (Fig. 220*a*) or made countersunk (Fig. 220*b*). Rivets with reinforced heads are shown in Fig. 220*c* and *h*.

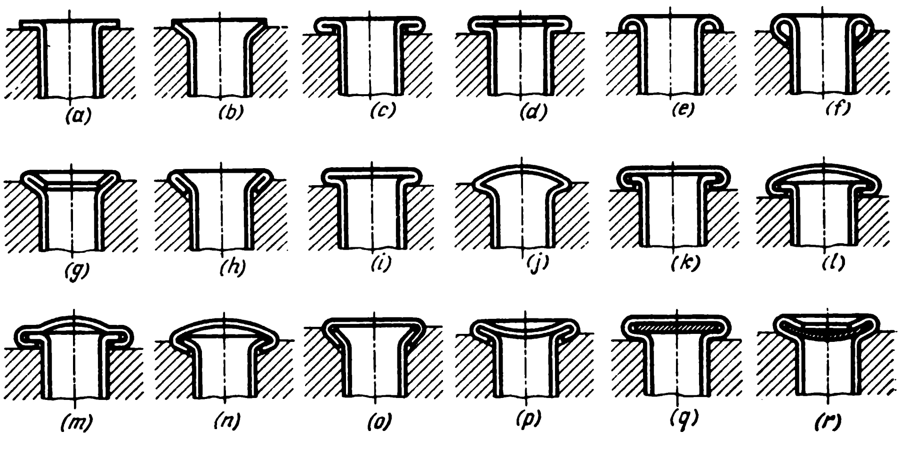


Fig. 220. Thin-walled tubular rivets

Such rivets mounted on the face surfaces of decorative parts are manufactured with solid heads formed by extrusion (Fig. 220*i*, *j*) or with composite heads (Fig. 220*k*-*r*).

The closing heads are formed by the methods shown in Fig. 220*a*-*h*. The heads in Fig. 220*a* and *b* are formed in one operation. The other designs require two operations, or are formed with the aid of a double-action set punch.

6.13. Blind Rivets

When spatial structures are riveted it is frequently impossible to approach the riveting tool to form the closing head (for example, in the case of rivets driven into internal cavities). In such cases use is made of *set rivets* fitted in and closed up from one side. These are usually tubular rivets spreaded with a set punch. The end of the shank is provided with a neck (Fig. 221*a*) or a conical step (Fig. 221*c*). During the clinching process the set punch expands the metal and forms the closing head (Fig. 221*b*, *d*).

To reduce the spreading force the thickened end of the rivet is slotted crosswise (Fig. 221*e*, *f*).

Rivets with a *non-recoverable punch* (Fig. 221*g*-*l*) offer greater advantages as to strength and simplicity of operation. For the greater strength after the closing head is formed the punch is secured in the shank of the rivet by means of rifles (Fig. 221*g*, *h*), conical recesses (Fig. 221*i*, *j*) or grooves (Fig. 221*k*, *l*) filled by flowing material in a plastic state.

A considerable axial force is required to drive in rivets of this type and they can therefore be employed only in rigid thick-walled structures.

When thin-sheet members are joined, the sheets should be relieved of the riveting forces. This purpose is often served by pull-out sprea-

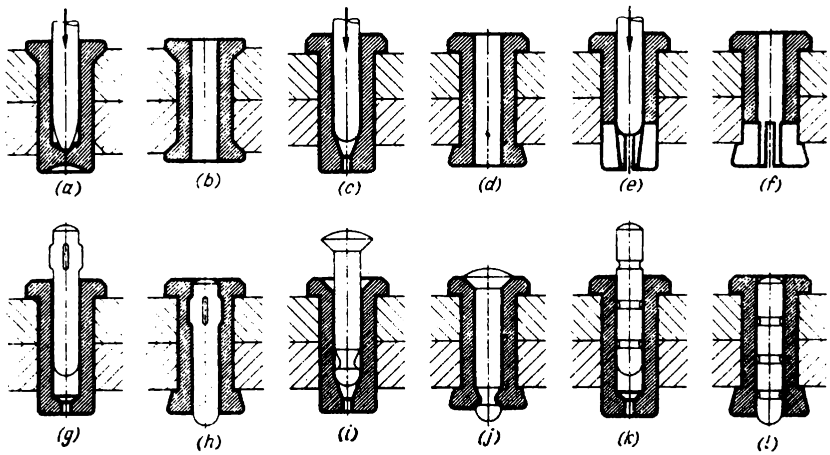


Fig. 221. Blind rivets

ders. The rivet with its head outside and fitted-in spreader is inserted into the hole. Then, resting against the head, the spreader is withdrawn forming the closing head.

Typical designs of light set rivets are illustrated in Fig. 222.

In Fig. 222, *1* the spreader is thickened at one end to a diameter exceeding that of the rivet hole. As the spreader is pulled out, it forms the closing head (Fig. 222, *2*) and at the same time expands the shank of the rivet fitting it tightly in the hole. Other modifications of this design are shown in Fig. 222, *3* and *4*.

In the design in Fig. 222, *5* the spreader has a cap connected to the stud by a thin neck *m*. During riveting the cap is enclosed in the head being formed (Fig. 222, *6*). After this the resistance to spreading sharply increases and the tool breaks at the thin portion. The cap remains in the head. In the design in Fig. 222, *7* the spreader built into the shank of the rivet is broken at the point *n*. The cap also remains in the head (Fig. 222, *8*).

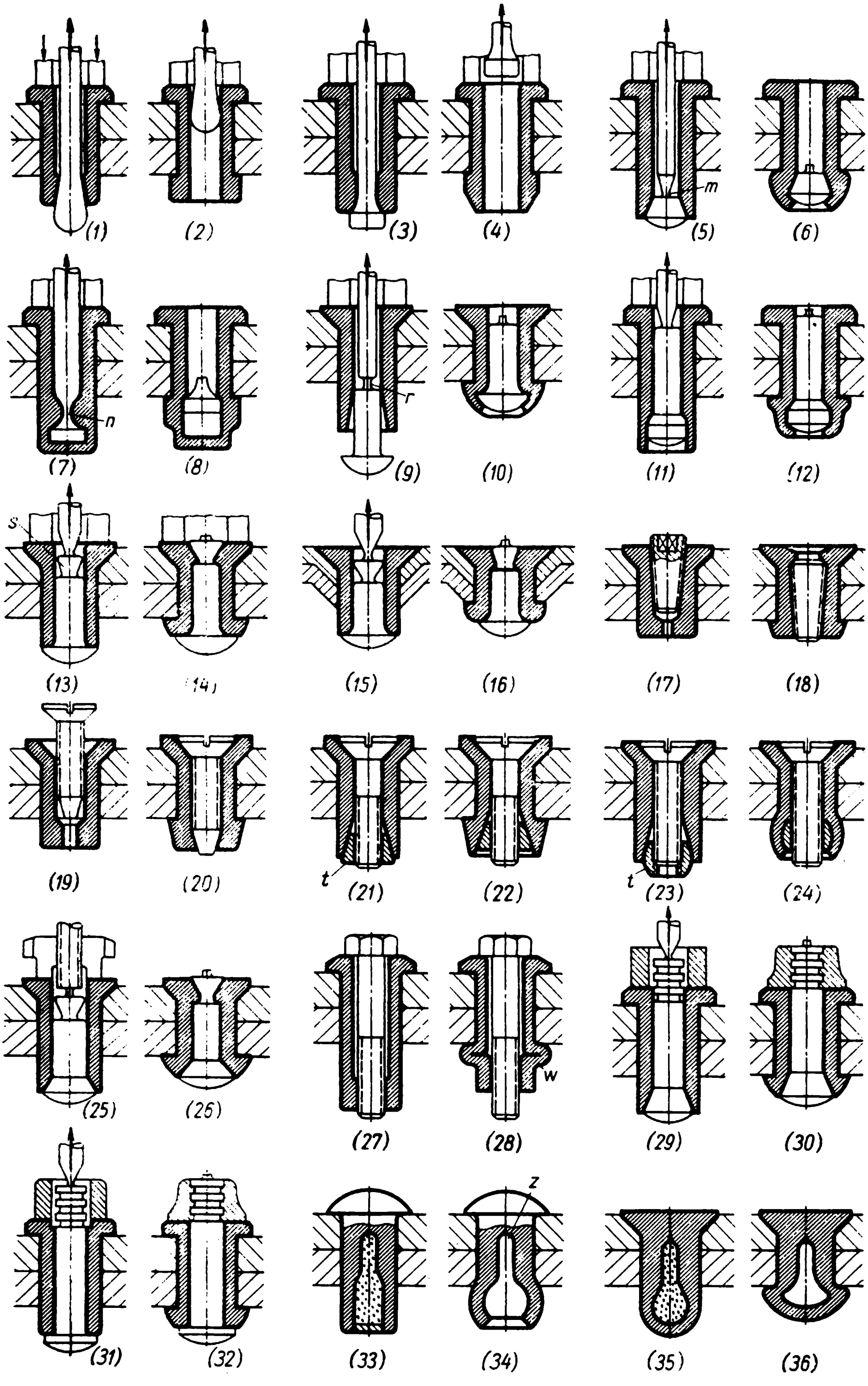


Fig. 222. Set rivets

Rivets with *non-recoverable spreaders* are stronger. In the design in Fig. 222, *9* the tool has a button-shaped end connected to the stud by a thin neck *r*. After the closing head is formed the neck is broken, the button is left in the rivet (Fig. 222, *10*) and is securely locked there. Modifications of this design are shown in Fig. 222, *11*, *12*.

In the designs in Fig. 222, *13*, *14* the detachable spreader is locked by inflow of the rivet material into cone *s*. Figure 222, *15*, *16* shows a rivet designed on the same principle used to join thin sheets.

The process of driving in the rivets illustrated in Fig. 222, *1-16* can be automatized. Today, highly efficient riveting machines are available with automatic orientation, feed and fitting in of rivets.

In Fig. 222, *17-20* the joined members are relieved of axial force by the use of screwed-in tapered (Fig. 222, *17*, *18*) and cylindrical (Fig. 222, *19*, *20*) punches. The punches are locked in the rivets by friction forces.

In Fig. 222, *21-24* the closing head is formed by a conical or spherical nut *t* pulled in by the rotation of a bolt. The designs in Fig. 222, *25*, *26* employ a stepped breaking spreader clamped by an external nut *v* and locked by inflow of the rivet material into the conical recess.

In Fig. 222, *27* the hole of the rivet is provided with a threaded portion into which a bolt resting against the rivet head is screwed in. When the bolt is turned, it pulls up the end of the rivet shank forming a thickened portion *w* used as the closing head (Fig. 222, *28*).

In Fig. 222, *17-28* the rivet should be secured against rotation during the initial tightening stage. These fitting methods are inferior to the previous ones.

Figure 222, *29-32* presents strong rivets closed up by means of a breaking spreader locked by cups made of plastic metal fitted into the annular grooves on the body of the tool. Such rivets are used, for example, in ship-building to connect massive platings.

The most productive and universal method is the one in which *explosive rivets* are employed. The shank of the rivet is filled with a charge (Fig. 222, *33*) which is exploded after the rivet is fitted (usually by applying an electrically heated holder onto the rivet head). The explosion forms the other end into a spherically shaped head (Fig. 222, *34*). The duct *z* is used to expand the shank in the abutting plane of the sheets being riveted.

Explosive rivets with an enclosed charge (Fig. 222, *35*, *36*) are more useful. The explosion occurs in the body of the rivet; the joint is relieved of the force of reaction of the gas jet; and the rivets can be fitted in noiselessly.

Explosive rivets made of aluminium alloy are widely employed to connect platings in the aircraft industry.

In strength, explosive rivets are inferior to other designs of set rivets (for example, to the load-bearing rivets in Fig. 222, *29-32*).

Set rivets are usually made with a diameter of 4-12 mm, and the rivets shown in Fig. 222, *29-32* with a diameter of up to 25 mm.

6.14. Special Rivets

Special *distance rivets* are employed to connect parts arranged at a preassigned distance from one another (Fig. 223*a*).

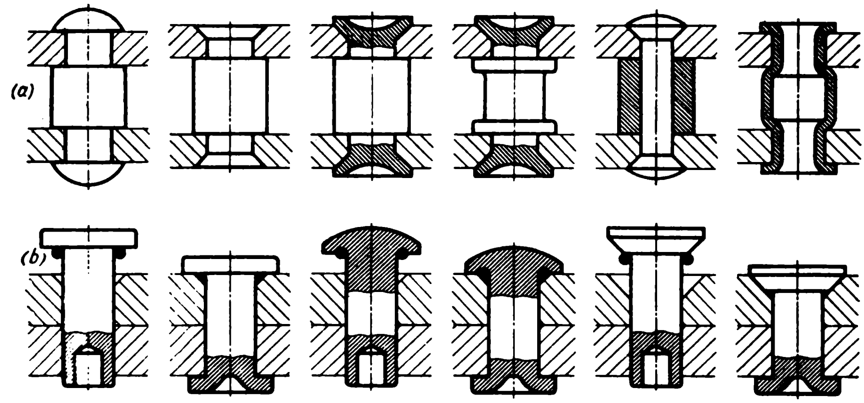


Fig. 223. Special rivets

In the case of hermetic rivets (Fig. 223*b*) a ring made of plastic metal (aluminium, annealed copper) or of elastomers (for joints operating at low temperatures) is placed under the head. When the rivet is driven in, the ring is flattened and seals the joint.

Tightness can also be attained by plating the rivets with cadmium or zinc.

6.15. Riveting of Thin Sheets

When thin sheets are attached to massive members the set head should be positioned on the side of the sheet, the closing head being formed on the side of the massive member (Fig. 224*b*). The design in Fig. 224*a* is wrong.

Countersunk heads on the side of the sheet are out of the question (Fig. 224*c*). To fit the sheet tightly the set head should have as large a diameter as possible (Fig. 224*d*) or a washer (Fig. 224*e*). It is good practice to make the washer of spring steel and slightly taper-like bent (Fig. 224*f*) to spread it up during clinching operation.

Figure 224*g-l* illustrates strengthened joints designs.

In Fig. 224*g* the edges of the holes in the sheet are flanged and are double folded in to form a strong joint when closing up the rivet (Fig. 224*h*).

In the design *i* in Fig. 224 with a countersunk head the edges of the hole under the head are pinched down during the riveting operation (Fig. 224*j*).

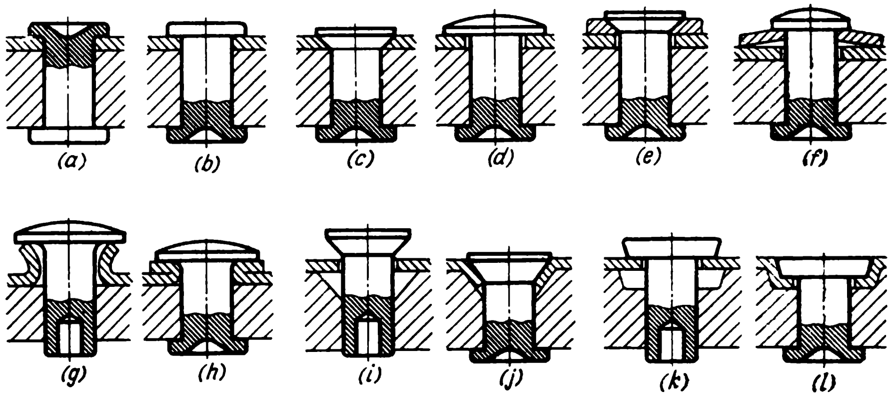


Fig. 224. Riveting of thin sheets

In Fig. 224*k* the sheet is formed during riveting by a tapered countersink (Fig. 224*l*).

Chapter 7

Fastening by Cold Plastic Deformation Methods

This way of fastening is employed in blind joints primarily to fix parts in relation to one another. In many cases such joints carry considerable loads.

Soft and ductile metals (relative elongation $\delta > 3$-4 per cent) can be deformed plastically in a cold state (for example, annealed steel, copper, aluminium and magnesium alloys, and annealed titanium alloys). As far as normalized and structurally improved steels are concerned plastic deformation is difficult. The methods of plastic deformation cannot be used for brittle metals (grey cast iron) and also for steel hardened or subjected to thermochemical treatment (carburizing, nitriding and cyaniding).

The principal methods of plastic deformation are as follows: clinching, expanding, spreading, calking and punching. Thin-sheet

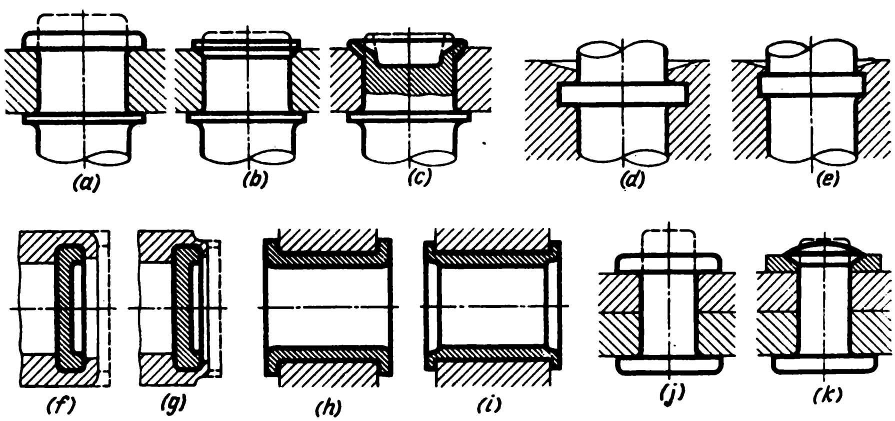


Fig. 225. Fastening by plastic deformation methods
a, d, f, h, j—irrational designs; *b, c, e, g, i, k*—rational designs

structures are also subjected to bending, outside flanging, beading and seaming.

As a rule, plastic deformation should be limited to a necessary minimum (Fig. 225). The smaller the volume of deformed metal and degree of deformation, the lesser the hazard of cracks and tears and the stronger the joint.

A reduction in the amount of plastic deformation lessens the force required for deformation, makes it possible to employ harder and stronger materials for the joints and increases, with all other conditions being equal, the efficiency of the fastening operations.

7.1. Fastening of Bushings

Figure 226 illustrates methods of fastening bushings by spreading the metal into taper seats (Fig. 226*a*) or into annular recesses in

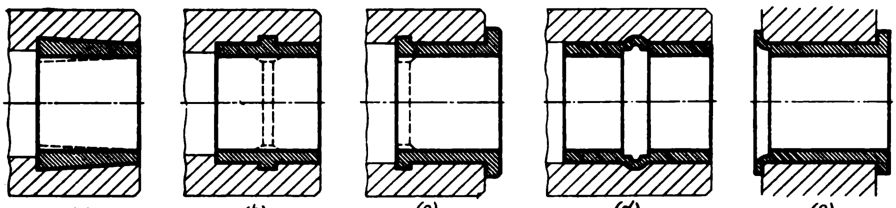


Fig. 226. Fastening of bushings

the locating holes (Fig. 226*b*, *c*). The laps (shown by the dash line) provided on the internal surface of the bushings are intended to allow the metal to expand.

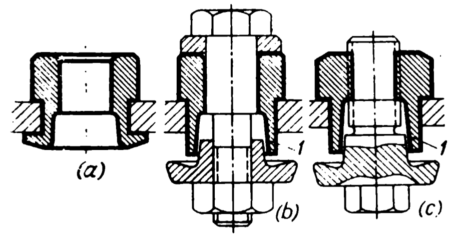


Fig. 227. Fastening of bushings in sheet members

The axial force of metal spreading is taken by the thrust of the end-face (Fig. 226*a*, *b*) or the flange (Fig. 226*c*) of the bushing against the enveloping part. After spreading the internal surface of the bushing is finish-machined by a sizing mandrel or a cutting tool.

Figure 226*d*, *e* shows the ways bushings are fastened by the expanding method.

Bushings are fastened in sheet members by expansion (Fig. 227*a*) or by spreading the protruding collar *1* under a press or with the aid of other appliances (Fig. 227*b*, *c*).

7.2. Fastening of Bars

Massive cylindrical parts (columns, pillars, etc.) are fastened by expanding their end-faces (Fig. 228).

The parts are usually set up with transition or interference fits. A heavy drive fit ensures the best joint, expansion serving only for additional safety.

Figure 229 shows the principal modes of fastening tubular bars in massive workpieces:

(1) the bar is inserted in an inverted tapered seat (Fig. 229*a*) and is fastened by expanding laps *1* with a cylindrical spreader (Fig. 229*b*);

(2) the bar is locked by expanding the massive end into the cylindrical slot in the seat (Fig. 229*c*, *d*);

(3) the bar is driven into a seat having a tapered insert. Bearing against shoulder *2* the bar is forced down to expand the bar material into the cylindrical slots of the seat (Fig. 229*e*, *f*).

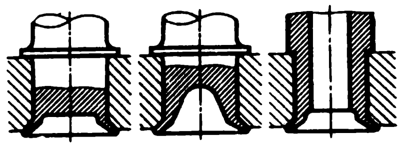


Fig. 228. Fastening of columns

In an improved design the taper made integral with the bar (Fig. 229*g*) is connected with the latter by a thin neck. When the bar is upset the neck breaks, the taper expands the end of the bar which is secured in the hole by annular rifles (Fig. 229*h*).

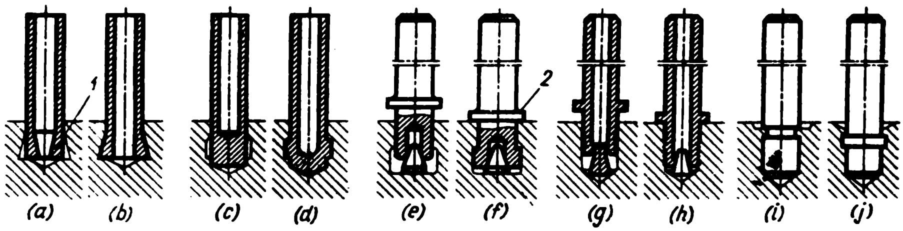


Fig. 229. Fastening of bars

Bars made of a material which does not yield to plastic deformation and fitted into soft metal workpieces are secured by calking the workpiece material into the annular groove (Fig. 229*i*) or onto the shoulder (Fig. 229*j*) of the bar.

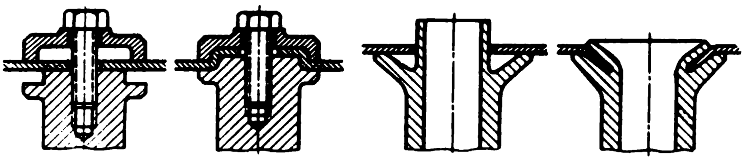


Fig. 230. Fastening of tubular bars to sheet members

Figure 230 illustrates methods of joining tubular bars to sheet members with plastic deformation of the sheet to increase the strength and rigidity of fastening.

7.3. Fastening of Axles and Pins

Pins made of soft material that yields to plastic deformation are fastened by clinching and expanding their ends (Fig. 231*a-c*), punching the pin ends at several points (Fig. 231*d*, *e*) and extrusion on the periphery of the pin end formed by an annular calking tool

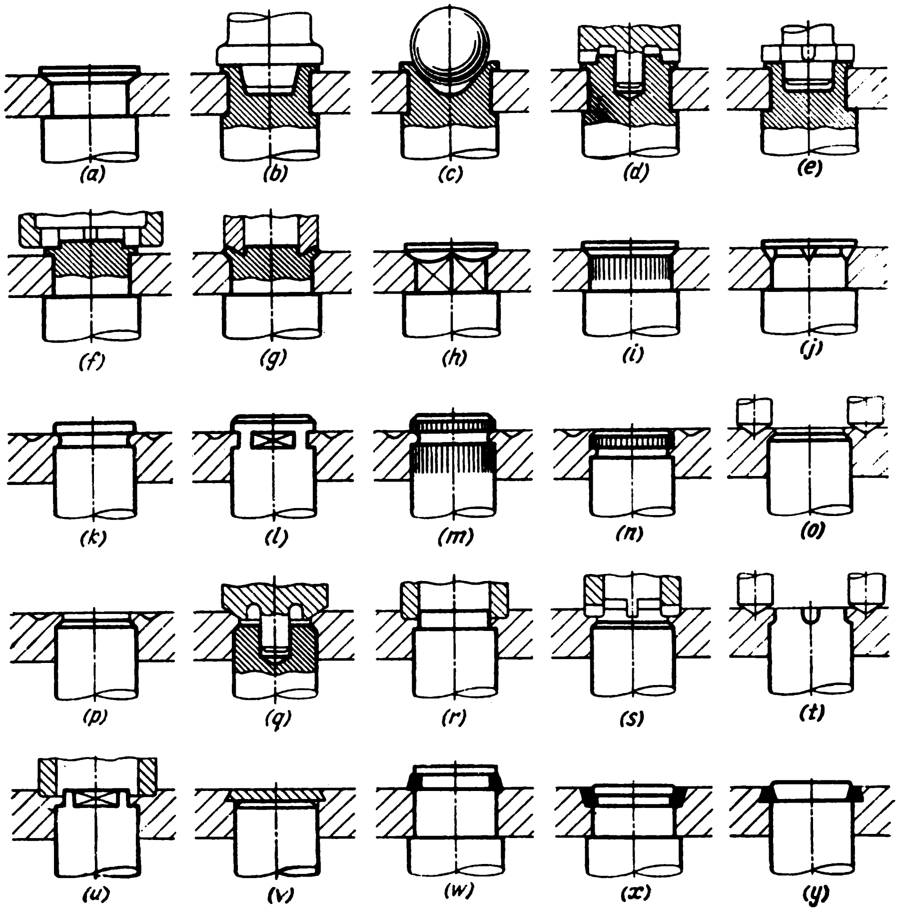


Fig. 231. Fastening of axles and pins in plates

with internal teeth (Fig. 231*f*). Figure 231*g* shows the method of fastening by expanding the end of the pin to a taper with an annular calking tool.

Figure 231*h-j* shows the methods of fastening when the pin is secured against rotation. Figure 231*h* shows the pin being locked by upsetting its square end in a square recess, and in Fig. 231*i* the pin is locked by fitting in its rifled end.

Figure 231*j* shows the simplest method when the cylindrical end of the pin is calked into triangular slots made in the chamfer of the seating hole. All-over clinching may be replaced by local deformation as shown in Fig. 231*d*, *e*.

Pins made of hard materials that cannot be clinched are fastened in plastic metal workpieces by forcing the workpiece material into the circular groove on the pin (Fig. 231*k*), calking the workpiece material into the flats on the pin (Fig. 231*l*) and fitting with the use of rifles (Fig. 231*m*, *n*).

Figure 231*n*-*u* illustrates the methods of fastening the pins in rigidly interconnected parts (for example, in the cheeks of forks, in shackles, etc.) when the pin is calked on both ends.

In Fig. 231*o* the pin is secured by punching the part at several points on its periphery, in Fig. 231*p*-*r* by circular expansion and in Fig. 231*s* by local extrusion.

Rotation of the pins is prevented by punching the metal of the part into the slots milled in the pin (Fig. 231*t*) or calking the material into flats on the pin (Fig. 231*u*).

If the pin and the cheeks are made of hard materials that cannot be clinched fastening is done by means of flattened plugs (Fig. 231*v*) or rings made of plastic materials (low-carbon steel, annealed copper, etc.) which are calked into the cuts in the pin (Fig. 231*w*-*y*).

7.4. Connection of Cylindrical Members

Coaxial cylindrical parts (for instance, bars and enveloping bushings) are joined by calking or rolling the bushing around annular shoulders (Fig. 232*a*) or into grooves (Fig. 232*b*, *c*) in the bar.

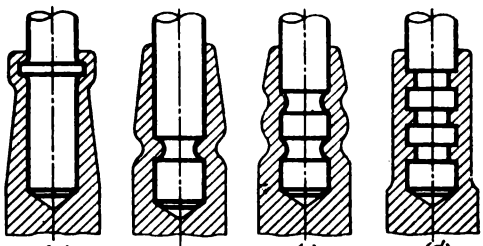


Fig. 232. Connection of cylindrical members

If, according to its function a joint requires free rotation of one element in relation to the other the surfaces to be connected are coated with a layer of separating graphite grease before calking. In such cases the grooves should be of rectangular shape (Fig. 232*d*).

7.5. Fastening of Parts on Surfaces

Small cylindrical parts such as bosses, contacts, supporting feet, etc., mounted on the surfaces of members are secured by calking in inverted cone seats (Fig. 233*a*-*f*).

The same methods are employed to fasten circular elements, for example, annular seals, valve seats etc. (Fig. 233*g*-*l*).

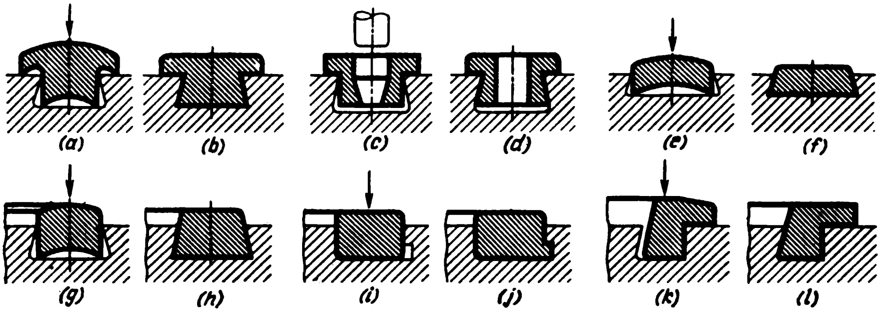


Fig. 233. Fastening of parts on surfaces

Figure 234 shows some methods of fastening valve seats.

Desings *a* and *b* (Fig. 234) are used for seats made of plastic metal (bronze, austenite steel, etc.) fitted into hard and brittle (cast

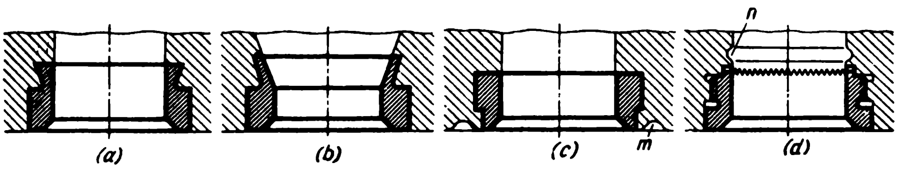


Fig. 234. Fastening of valve seats

iron) metal housings, and designs *c* and *d* (Fig. 234) for seats manufactured from hard metal and inserted into plastic metal housings (aluminium alloy).

In Fig. 234*c*, fastening is done by calking or rolling the material of the housing around the seat (section *m*).

In Fig. 234*d*, the seat is screwed into the housing and locked in position by rolling the circular groove in the hole (section *n*) with the following metal inflow into teeth cut in the underneath end-face of the seat.

Segments, flat springs and similar parts are secured to the surface of large parts by fitting them into slots (Fig. 235*a*) and spreading

the material with a punch at several points. Longitudinal movement of the segment is prevented by filling semicircular cuts with metal.

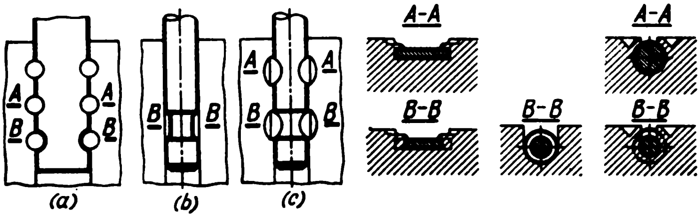


Fig. 235. Fastening of segments and rods

A similar method is also used to secure cylindrical rod-type parts (Fig. 235*b*, *c*).

7.6. Swaging Down of Annular Parts on Shafts

The method of plastic deformation is frequently employed to calk cylindrical elements such as rings (Fig. 236*a*, *b*) and sleeves (Fig. 236*c*) on shafts.

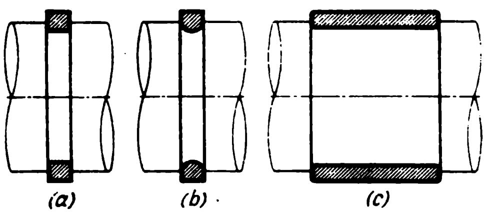


Fig. 236. Swaging down of rings and sleeves on shafts

Parts of this type are swaged by presses with split bushings or still better on rotary swaging machines applying the effort simultaneously at several points on the periphery and smoothly increasing the force. Hand calking and all-round rolling cannot be used in this case since they spread the bushing instead of giving it the necessary compression.

7.7. Fastening of Plugs

Plugs are secured in hollow shafts by expanding the shaft (Fig. 237*a*, *b*), by calking the plug periphery until the metal fills the annular groove first made in the shaft (Fig. 237*c*, *d*) and by spreading the plug rim with a tapered punch (Fig. 237*e*).

Figure 237*f*, *g* illustrates the methods of fastening thin-sheet plugs by expanding them into an annular slot in the hole of the shaft.

Flattening of plugs is also extensively employed (Fig. 237*h*). Initially, the plug has a spherical form and is spread by a flat punch

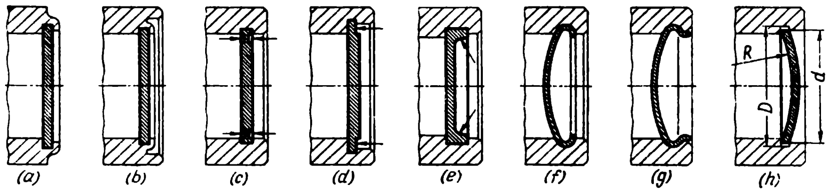


Fig. 237. Fastening of plugs

with the other side resting against a flat support. The plug periphery is then forced in the shaft groove.

Radius R of the sphere can be found from the formula

$$R = \frac{0.25d}{\sqrt{\left(\frac{D}{d}\right)^2 - 1}}$$

where d = plug diameter
D = diameter of the groove for the plug

On average $\frac{D}{d} = 1.03$ in which case $R \approx d$.

Figure 238 shows the methods of fitting plugs into thin-walled pipes.

The joint should be designed so that supports may be used to form and flatten the seam: flat ones (over surface m in Fig. 238*a*-*c*)

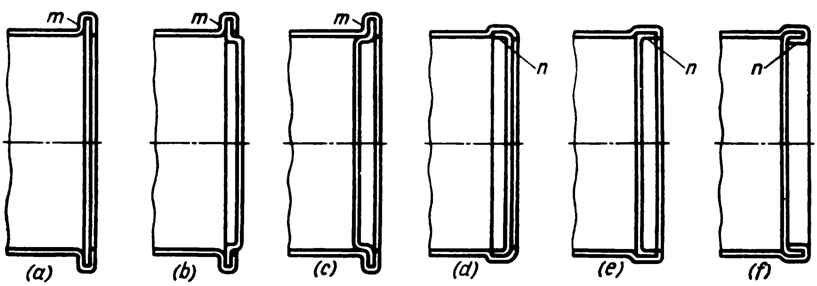


Fig. 238. Fastening of plugs in pipes

or cylindrical ones (over surface n in Fig. 238*d*-*f*). The designs that allow the admission of cylindrical supports from outside (Fig. 238*e*, *f*) are preferable to the ones where an internal support is necessary (Fig. 238*d*).

7.8. Fastening of Flanges to Pipes

Flanges are attached to thick-walled pipes (walls 4-6 mm thick) by rolling the pipe ends into annular grooves in the flanges (Fig. 239*a*, *b*).

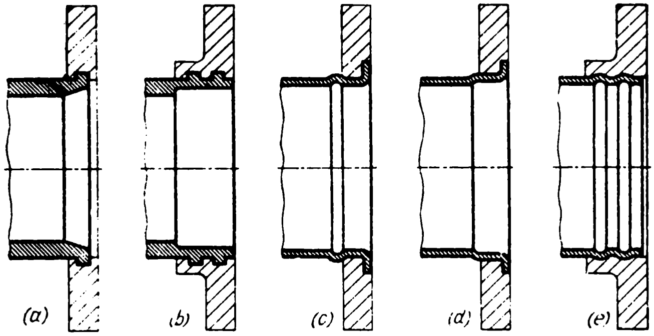


Fig. 239. Fastening of flanges to pipes

The methods of fastening flanges to thin-walled pipes are illustrated in Fig. 239*c*-*e*.

7.9. Fastening of Tubes

Figure 240*a*-*c* shows the methods of fastening tubes in sheets and plates. Thick-walled tubes with a wall thickness of 2-5 mm

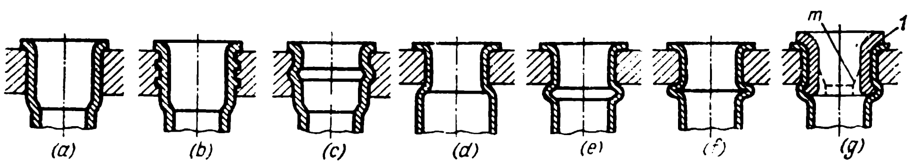


Fig. 240. Fastening of tubes in sheets and plates

(fire and water tubes of boilers) are fastened by expanding the ends of the tubes by rollers. Annular grooves (Fig. 240*b*, *c*) are provided in the hole to increase the strength and tightness of the joint.

Figure 240*d*-*g* presents methods of fastening thin-walled tubes. Fig. 240*g* shows the strongest and stiffest design. In this design the tube is secured by a thick bushing *1* with a tapered projection *m* expanded when the tube is installed.

Figure 241 shows methods of fastening oil-feed tubes in shafts, drawn (Fig. 241*a*-*f*) and turned (Fig. 241*g*-*i*).

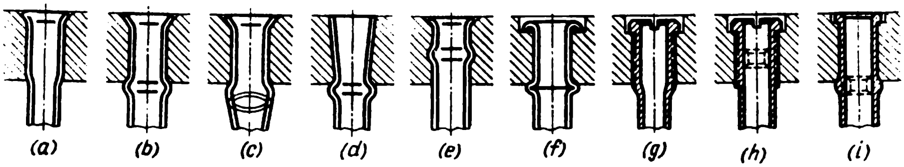


Fig. 241. Fastening of oil-feed tubes in shafts

7.10. Fastening by Means of Lugs

This mode of connection is used for thin-sheet structures. The lugs (Fig. 242*a*-*c*) on one of the parts to be connected are introduced into the slits in the adjacent part and bent over.

Another method is flanging the lugs perpendicular to their plane (Fig. 242*d*). This method is used when the design does not allow a bending force to be applied.

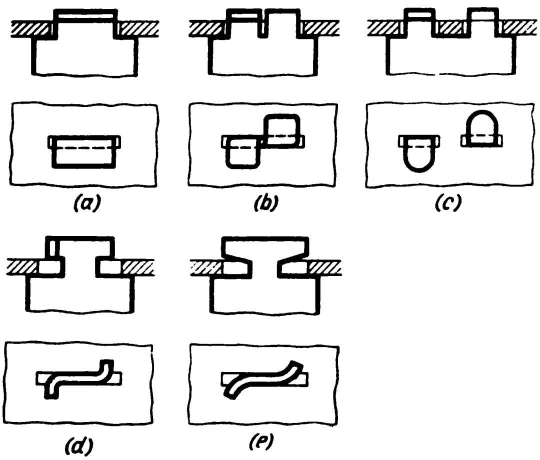


Fig. 242. Fastening by means of lugs

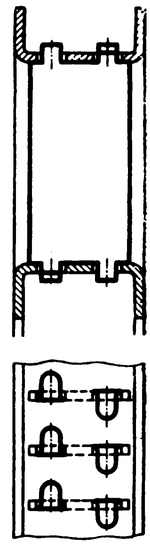


Fig. 243. Fastening blades of a return-circuit rig

The joint can be tightened with a certain interference if a chamfered cut is used (Fig. 242*e*).

The strength of such joints is not high. In some cases these methods are employed in power structures. Figure 243 shows a unit for fastening blades to the shells of an annular return-circuit rig of an axial air compressor. The large number of attachment points makes this design sufficiently strong and rigid.

7.11. Various Connections

Figure 244 illustrates the methods of connecting sheets and plates by fluting them with punch *1* in thin sheet (Fig. 244*a*) or in thick plate (Fig. 244*b*), the closing head being formed by die *2* installed on the side opposite to the motion of the punch.

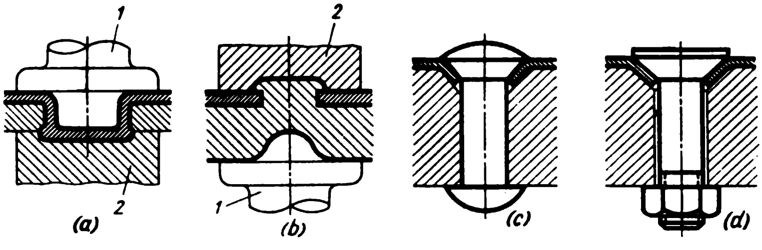


Fig. 244. Fastening of lining sheets

The fastening of lining sheets to massive members by rivets (Fig. 244*c*) or bolts (Fig. 244*d*) can be reinforced by plastic deformation of the lining at the joint.

Figure 245*a*, *b* illustrates a method of spreading the shaft material into tapered recesses in disks when mounting light gears and

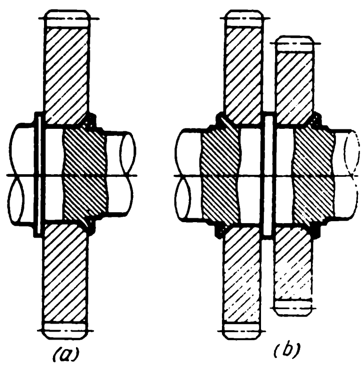


Fig 245. Fastening of disk-shaped parts on shafts

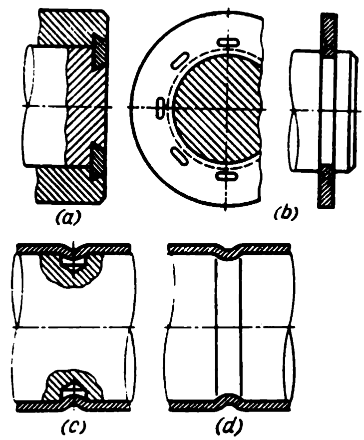


Fig. 246. Units connected by plastic deformation methods

other disk-shaped parts on stepped shafts. Figure 246 shows: fastening of a hub on a shaft by means of a flattened washer (Fig. 246*a*), fastening of a disk by calking the metal into an annular recess in a shaft (Fig. 246*b*), fastening of a sleeve on a shaft by calking the metal into recesses (Fig. 246*c*) or into an annular groove (Fig. 246*d*) in a shaft.

7.12. Seaming

Seaming is used to connect sheet workpieces of various thicknesses ranging from several tenths of a millimetre (tin) to 1-2 mm.

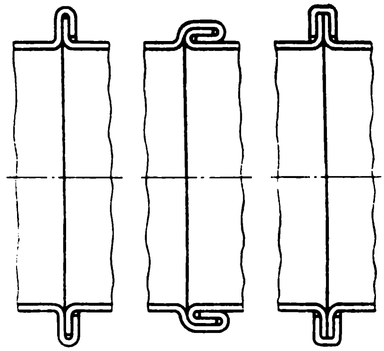


Fig. 247. Seaming of pipes

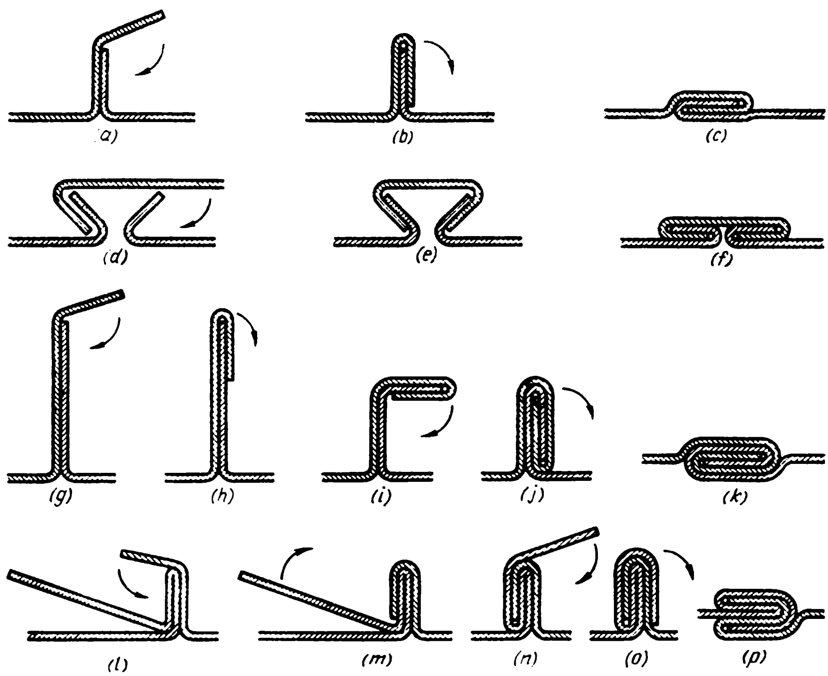


Fig. 248. Seam joints

Figure 247 shows seam joints used to connect thin-walled pipes and shells.

The most common joint is the one in which the edges are first flanged (Fig. 248*a*) to form a lock (Fig. 248*b*) after which the lock is bent over and flattened to make a four-layer seam (Fig. 248*c*).

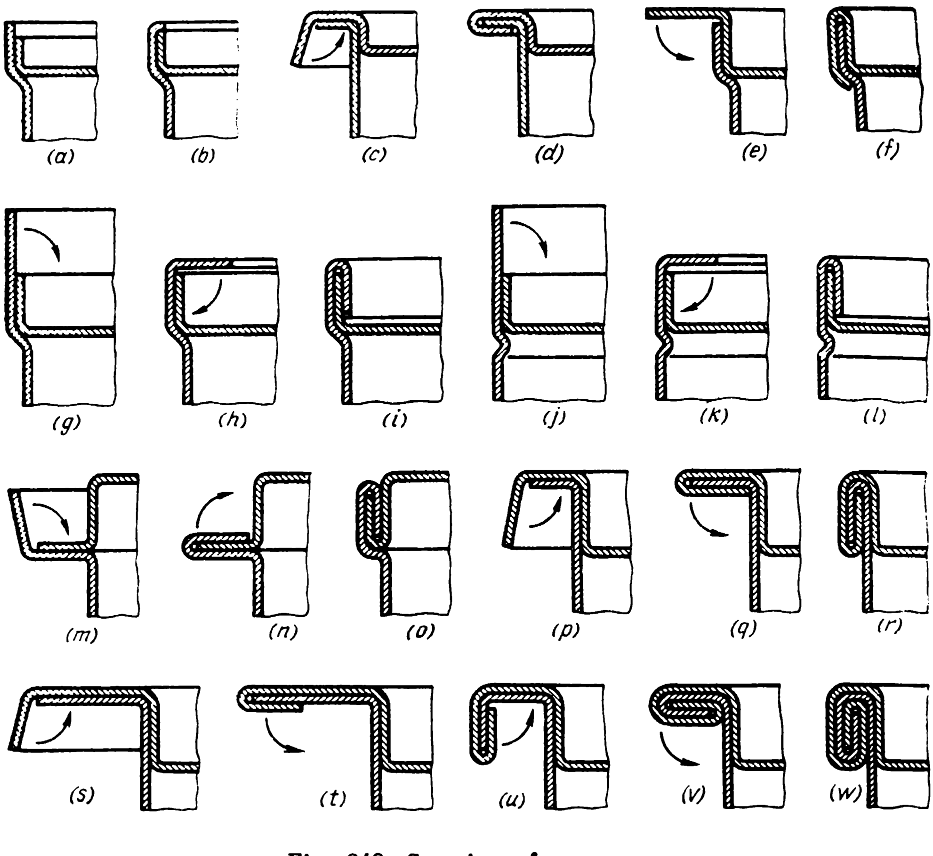


Fig. 249. Seaming of covers

Figure 248*d-f* presents a seam overlapped by a strip of sheet material, Fig. 248*g-k*—a strengthened six-layer seam (combination seam) and Fig. 248*l-p*—a seven-layer seam.

The seam joints shown in Fig. 248 are employed to connect flat sheets and form longitudinal seams of cylindrical shells.

Figure 249 shows methods of seaming bottoms and covers to cylindrical shells.

Figure 249*a*, *b* shows designs employed to connect comparatively thick materials (0.5-2 mm).

Tin products are connected by three-layer (Fig. 249*c-l*), four-layer (Fig. 249*m-o*), five-layer (Fig. 249*p-r*) or seven-layer (Fig. 249*s-w*)

seams. The seam is flattened during the last operation by pressing it against a centre mandrel placed in the bottom recess.

The most popular types of seaming are shown in Fig. 249*p-r*.

Figure 250 illustrates mechanized seaming of such joints on multiple-position rotor machines. Seaming is done in chucks consisting of a centre mandrel *1* and rollers *2* and *3* performing planetary motion around the workpiece.

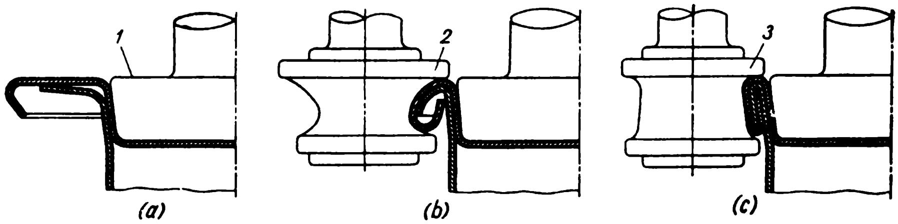


Fig. 250. Diagrams of machine seaming

Ordinarily, use is made of two rollers arranged diametrally on the periphery.

The initial stage of the operation is shown in Fig. 250*a*. The cover is delivered for seaming with its edges already bent. The edges of the shell are also flanged in advance.

At first, rollers *2* used for the first operation are brought to the workpiece (Fig. 250*b*) to form the seam, and then rollers *3* for the second operation (Fig. 250*c*) to flatten and compact the seam.

The rollers for the first and second operations are usually mounted in a staggered order in one chuck. As the rotor revolves the rollers for the first operation and then the rollers for the second operation are automatically brought into use.

Multiple-chuck seaming machines operating on this principle can finish up to 500 pieces per minute.

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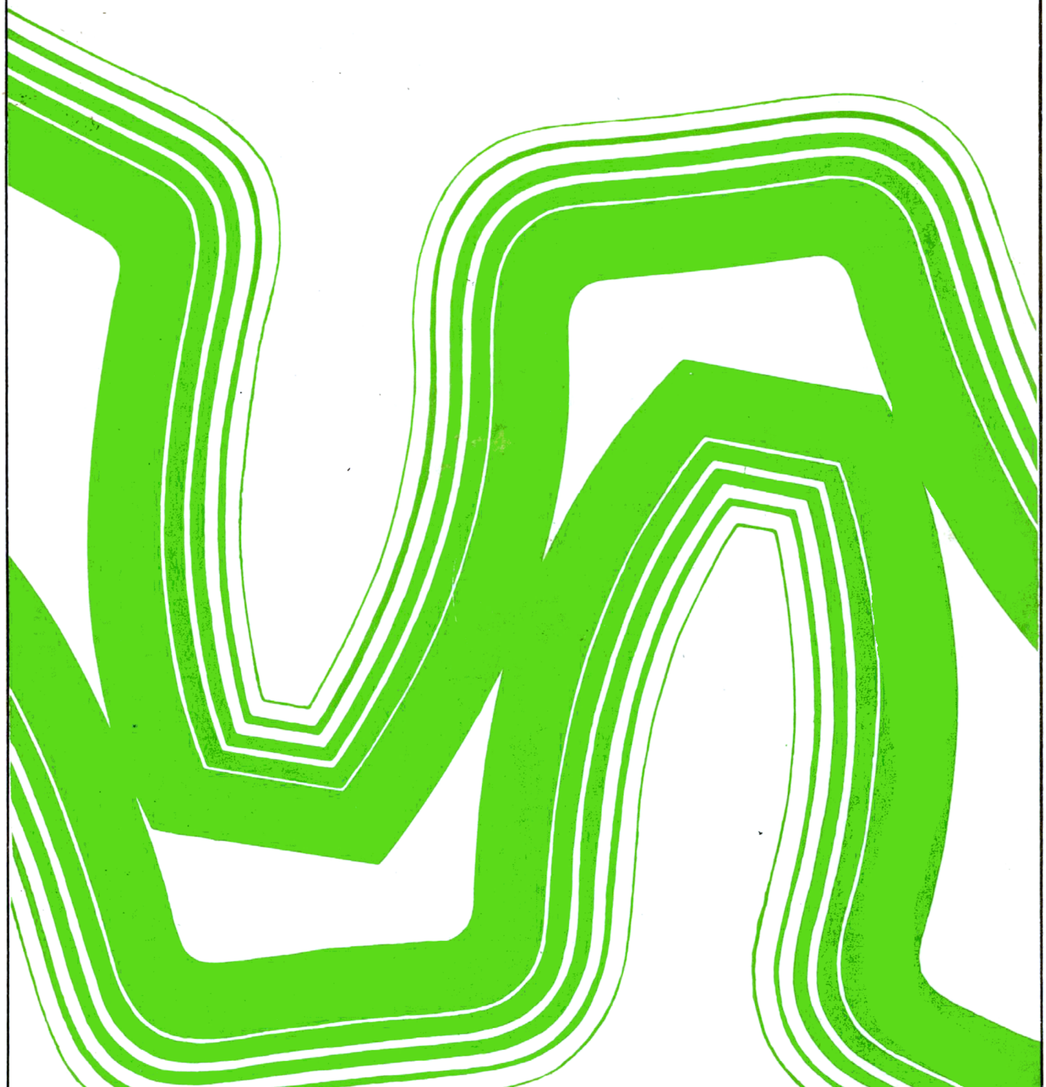

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